POWER

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A Parable

THERE was a land that knew but little rain, but through the canyon that divided it, so deep down and so far away that it looked like a silver thread, there ran a mighty river.

And all day long and all their lives men clambered down the rocky trail and climbed again the steep, high canyon wall bearing water to supply the people and their cattle and to moisten the parched earth that it might yield a harvest.

And the Chiefs and the Elders who furnished the vessels that they used and who owned the reservoirs and controlled the distribution and the access to the river, gave to each of them for each trip that he made, a metal token which might be exchanged for some of the water that he had brought or for other of his necessities.

And they grew and thrived and became a great people, so that there were more of them than were needed to till the ground and tend the herds and carry the water.

And the leaders among the workers spake unto their fellows and counseled them, saying, "Fill not your vessels so full and be not over mindful that ye spill not some upon the way, lest by too generous a service we deprive some of our number of a chance to earn.

"And be not overcareful of the vessels and the tools and conduits of our masters, for verily each of these that is lost or broken gives need for the labor of our brothers who fashion them.

"Moreover, why should we, without whose toil this land would be as nought, support the heat and burden of the trip for a paltry token, while they who pay us toil not, but live in ease and luxury?"

And they went to the Chiefs and Elders and in a mighty voice demanded that their wage be doubled And they, being thus compelled, submitted and thereafter gave to each two tokens instead of one.

But to those who bought the water they charged also double the number of tokens; whereat, the tillers of the soil, the spinners and the weavers and craftsman of all kinds, finding the purchasing power of their tokens thus reduced, increased the prices of their products and their service in the same degree.

And so it came to pass that all men received twice the number of tokens as in the beginning—and labored just as long and just as wearisomely, and ate and drank and lived no better than before, but gloried in their princely wage.

And he who had in the olden days denied himself and laid away of his tokens a portion that they might be to him a staff and support for his declining years, found that one-half of their value had departed.

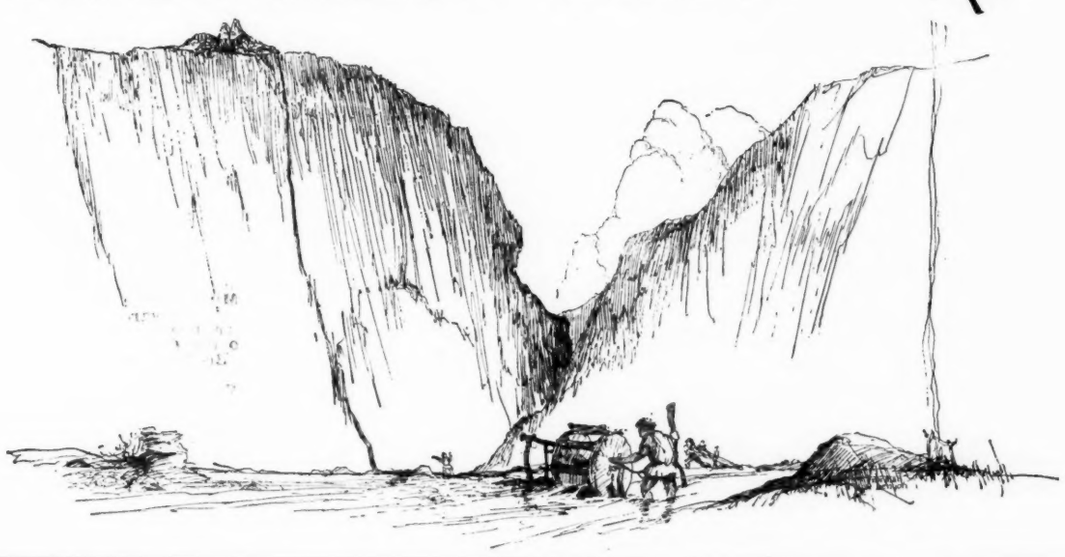
And one day as one of the toilers stood beside the river and sensed the power of the rushing flood, there came to him a vision. And he saw that energy controlled, the rushing waters made to lift a portion of themselves up the steep cliff, a copiously watered plain, a people relieved of the major portion of their burden.

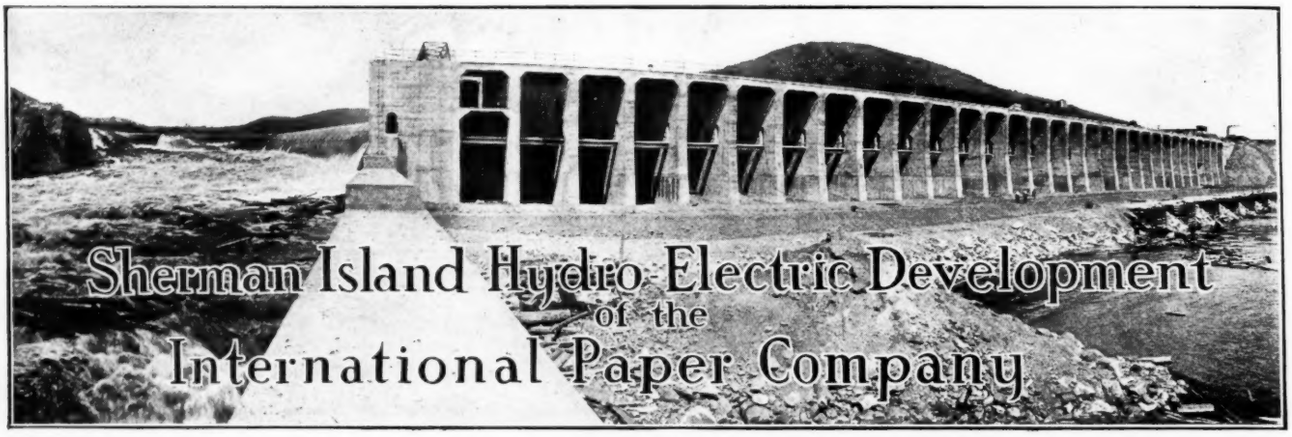
And with inspired genius he wrought out the details of his plan, until, complete and demonstrated in every particular, he displayed it to the Chiefs and Elders and to those whose burden he would remove.

But with one voice they cried, "Away with him! Crucify him! He would deny us that we should labor and earn our daily bread. It is an abomination to the Lord that water should run up hill."

And they spat upon him and despised him and kept on to labor and to complain the one against the other as was their wont.

J. R. Low





Sherman Island Hydro-Electric Development of the International Paper Company

MAIN DAM AND SPILLWAY. DAM RESTS ON THE GLACIAL DEPOSITS WHICH FORM THE RIVER'S BED

A FIFTY THOUSAND-HP. development on the Hudson River, consisting of five 10,000-hp. vertical Francis turbines to operate under 66-ft., head. The power house and canal are constructed on sand underlain in many places with quicksand. The dam rests on glacial deposits and is of special construction and the canal, over 3,500 ft. long, is concrete lined. Structural-Steel framework used to reinforce draft-tube bells. Provisions are made for continuously measuring the flow of water to each unit.

At Glens Falls, N. Y., and above on the Hudson River the International Paper Co. has water rights at a number of power sites, many of which have been developed for operation of the company's paper mills. In accordance with the general trend throughout the country to develop water power, the company is developing the remaining sites just as rapidly as circumstances permit. One of the most important of these power sites is that at Sherman Island, about seven miles above Glens Falls. The Lake Champlain canal feeder dam about three miles above Glens Falls backs the water up the river approximately four miles to Sherman Island. From this island to the Spiers Falls power plant of the Adirondack Power & Light Co. there is a fall in the river of about 67 ft. with a maximum unregulated flow of 100,000 cu.ft.-sec. and a minimum of 1,000 cu.ft.-sec. The work of developing this site is about finished and it will have an ultimate capacity of 50,000 hp. under a normal head of 66 ft.

At present there are being installed only four 10,000-hp. units, each requiring at full load about 1,520 cu.ft.-sec. The fifth unit will be installed when storage reservoirs are provided to control the river's flow. A portion of the power from these plants will be stepped up to 66,000 volts and sold to the Adirondack Power & Light Co.

Three of the outstanding features of the development are that practically every part of it is built on sand and gravel foundations, whereas most developments of this kind have solid rock, the special construction of the draft tubes and the methods provided for obtaining complete records of the water flow to the plant and turbines.

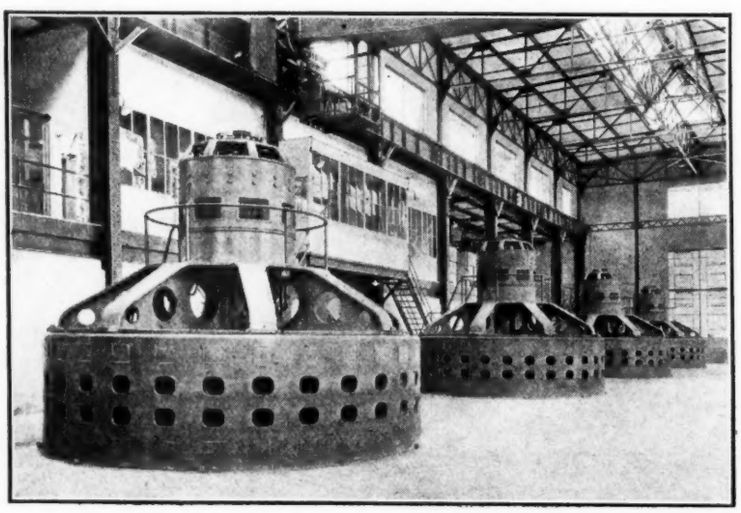


FIG. 1—INTERIOR OF POWER HOUSE SHOWS THE FOUR 10,000-HP. UNITS NOW INSTALLED

At the location of the development the river's banks and bed are glacial deposits of great depth, consisting chiefly of sand and gravel and in some places quicksand. Fig. 2 shows the general arrangement of the development. The site chosen for the dam is on the edge of a valley between the steep granite slopes of two peaks that flank the river and dip sharply under the glacial deposits. This provided rock anchorage at the two ends of the dam, although in the bed of the river it rests on glacial deposits of gravel and bolders. Between the dam site and the upper end of the canal-feeder pool formed by the dam above Glens Falls, there is a rapids that falls about 19 ft. To utilize this head would require excavating a long tailrace in the river's bed if the power house was located at the dam. This scheme was abandoned for a headrace canal along the river's bank from the dam to below the rapids, as the formation of the bank lent itself to such a development.

The main dam (see headpiece) across the river is 551 ft. long and has a maximum height of 70 ft. It is built of reinforced concrete and is of the multiple-arch type. The buttresses supporting the arches

rest on a heavy reinforced floor slab which distributes the weight of the dam over the whole foundation area (see Fig. 4). On account of the uncertainty of the river bed the use of an overflow type of dam was decided against and the spillway channel cut out of the rock at the south end of the dam. The spillway dam, as indicated in Fig. 2, is of a horseshoe shape of the gravity type and 864 ft. long exclusive of a bulkhead section in which sluice gates are provided for. This length will take a flood of 100,000 cu.ft.-sec. with a margin of two feet below the top of the main dam. The bulkhead section contains seven sluice gates that may be opened to partly empty the pond should it become necessary.

Between the north end of the main dam and the canal headworks there is a wing dam 171 ft. long of gravity section, built on solid rock and forming the east abutment of the canal headworks. The canal headworks is on the solid rock channel that forms the upper end of the canal, is designed as a dam to sustain a head of 35 ft., is constructed of reinforced concrete and contains ten gates made in three sections each,

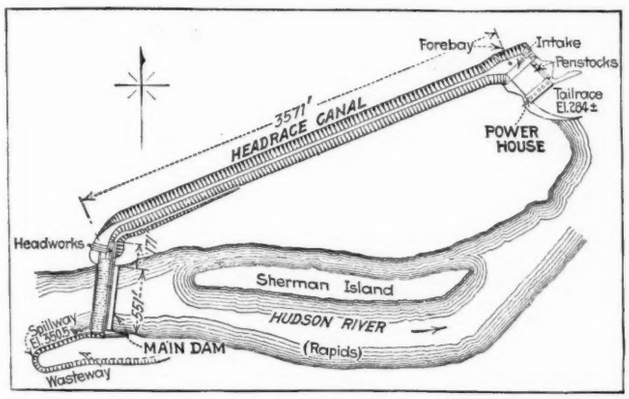


FIG. 2—GENERAL ARRANGEMENT OF DEVELOPMENT

7 ft. by 10 ft. 2 in. The gates are of reinforced concrete with angle iron to protect the edge and make tight-fitting joints. The sections are 18 in. thick and are fastened together with manganese-bronze links and pins. The links between the upper and middle sections are slotted to provide a 12-in. opening for filling the canal before removing the lower sections. As the gates are seldom used, it was desirable to make them as simple as possible at minimum expense, and reinforced concrete fulfilled these requirements. When not in use they will be removed entirely by a 25-ton electrically operated hoist. The gates weigh 27 tons, and with a full head of water against them it would have required a much stronger crane than that provided. This difficulty was overcome by making the links between the

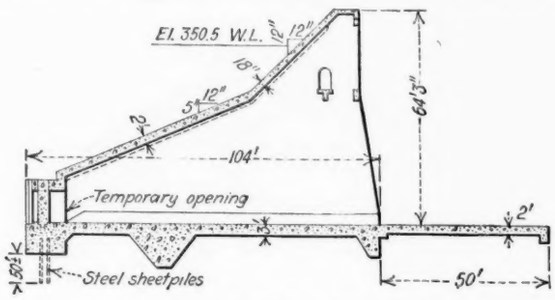


FIG. 4—CROSS-SECTION OF MAIN DAM

middle and upper sections of the gates with slots to allow lifting the top section 15 in., after which it is allowed to drop back 3 in., where it is held by latches catching into pockets in the concrete columns between the gates. This allows a 12-in. opening through the gates for filling the canal. After the canal is filled, only the dead weight of the gates in water has to be lifted which can be easily done with the 25-ton crane. With the top section of one gate lifted, the canal was filled in eight hours, therefore with the ten open less than one hour will be required.

From the headworks at the main dam to the forebay above the power house, a distance of about 3,600 ft., the water is carried in a reinforced-concrete lined headrace canal of a form shown in Fig. 7. With the exception of the first one hundred feet on one side, the canal is excavated in a glacial deposit ranging from coarse gravel to quicksand, most of it through fine sand. For this reason it was concrete lined up to six feet above the spillway level. Above this level the slope is riprapped to protect against scouring during flood periods. Because the canal lining rests on sand, it was necessary to provide for some settling, and this has been done by using a copper expansion strip at 21-ft. intervals. This construction allows a great deal of flexibility with a negligible amount of leakage.

The canal terminates in a reinforced-concrete lined

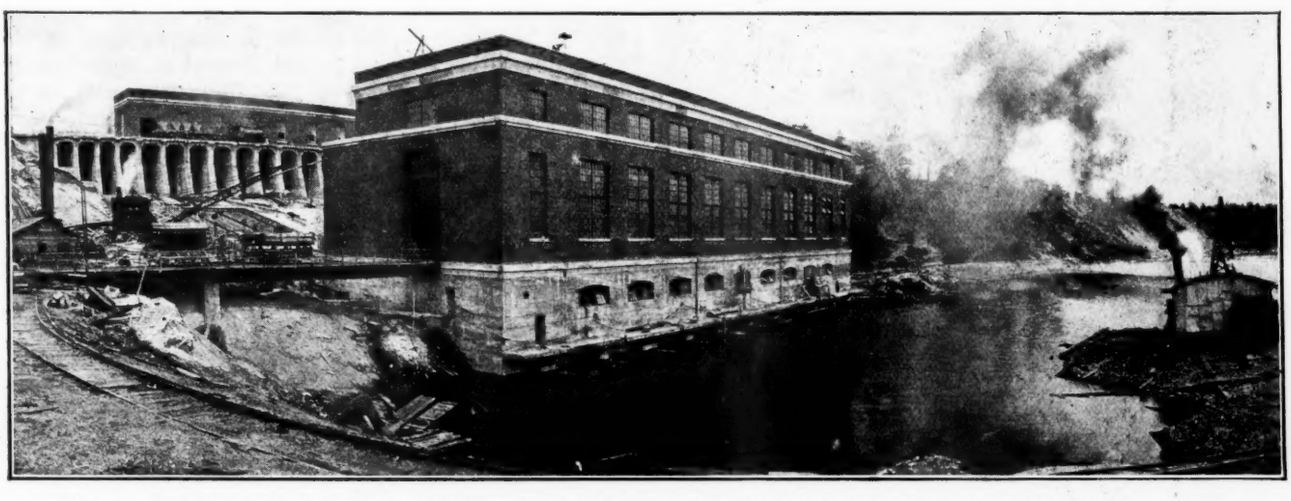
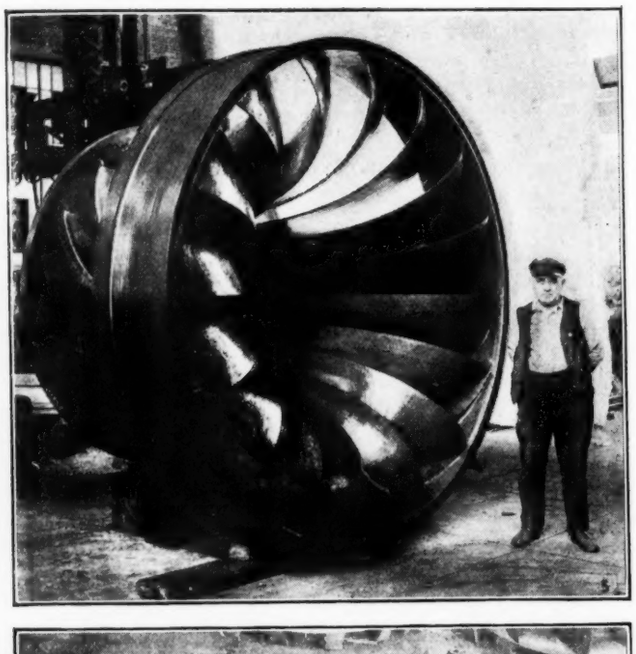
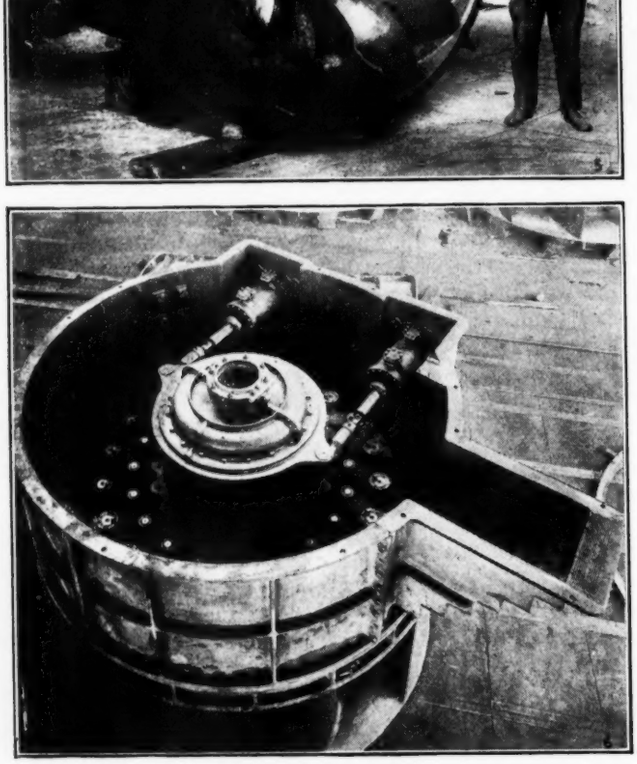


FIG. 3—THREE REINFORCED-CONCRETE PENSTOCKS PER UNIT CONNECT HEAD WORKS TO POWER HOUSE





forebay, which is lined up to above high-water level on account of the surges. A reinforced-concrete forebay dam forms the entrance to the penstocks. This dam is built on sand and rests on a heavy slab of reinforced concrete to distribute the load equally over the entire area occupied by the headworks. In these headworks are provided the entrances to the fifteen penstocks, three for each unit, and the two entrances to the ice and log chute. The two entrances to the log chute are closed by steel-framed concrete-covered gates, 8x12 ft. For a distance of about fifty feet below the headworks the log chute consists of two concrete tubes and then enters a wooden chute which runs into the river at the east end of the power plant.

To prevent the water from getting under the front of the headworks-foundation slab, it is provided with a steel sheet-pile cutoff. Overhanging the front of the forebay headworks is a curtain wall extending to about four feet below normal water level to prevent logs or large pieces of floating material reaching the racks. The racks are made in two sections of ⅜x4-in. iron on edge, assembled on through bolts with spacers between the bars. These racks rest on steel beams held in the concrete piers forming the divisions in the intake and are designed as a dam to take the full head of water.

At the top and just back of the racks is a trash trough running the full length of the headworks. This trough is given a 6-in. incline toward the ice and log chute. At the high end of the trough water can be admitted through a 12-in. circular valve, and at the low end entrance to the log chute is obtained through an 18x25-in. cast-iron valve.

Three Penstocks to Each Turbine

There are three penstocks to each turbine. Each one can be closed at its upper end with a Broome caterpillar-type sluice gate operated by an electric-motor-driven hoist and controlled by the operator at the switchboard in the power house. The piers between the entrance to each group of penstocks have been designed as a dam to withstand the full head of water. The two intermediate piers are of lighter construction, but a 2x4-ft. opening is provided in them so as to equalize the flow in case of obstruction on the racks or when putting in the stop-logs in front of the racks.

In the design of the penstocks, which are 176 ft. long, both reinforced concrete and steel were considered. In respect to cost both were about on a par. However, if steel had been used it would require a penstock diameter of about 19 ft., which would have involved structural difficulties both at the intake and at the entrance to the turbines. It would also have required extra excavation between the forebay and the plant. For these and other reasons reinforced-concrete penstocks were used, as this allowed using three small ones instead of one large one. The penstocks have been designed to cover practically the whole surface between the forebay headworks and the power house. This gave a light loading on the sand on which they rest.

FIG. 5—ONE OF THE 10,000-HP. TURBINE RUNNERS, WHICH OPERATE AT 150 R.P.M.; SPECIFIC SPEED 80.

FIG. 6—VIEW DOWN INTO CAST-IRON PIT LINER SHOWING GUIDE-VANE OPERATING GEAR AND HOW THE TWO SERVO MOTORS ARE SUPPORTED FROM THE SIDE OF THE PIT LINER.

FIG. 7—CANAL EXCAVATED IN FINE SAND AND LINED WITH REINFORCED CONCRETE WITH COPPER EXPANSION JOINTS, AT 21-FT. INTERVALS, WHICH MAKE IT PRACTICALLY WATERTIGHT.

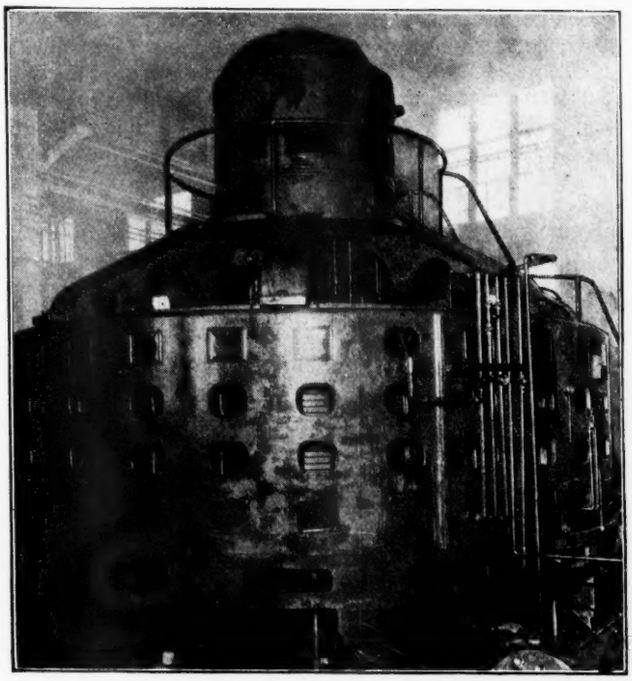
The penstocks are 10 ft. by 10 ft. 8 in. in section, are heavily reinforced and have a minimum wall thickness of 24 in. Each group of three serves one turbine, as shown in Figs. 12 and 14. As the whole penstock structure rests on sand, some of which is soft, it is divided into sections by transverse and longitudinal joints to provide flexibility. A tar construction joint is made between each group of six penstocks serving two turbines. There are also five traverse joints closed with lead strips and in addition they are calked on the inside with tar and oakum.

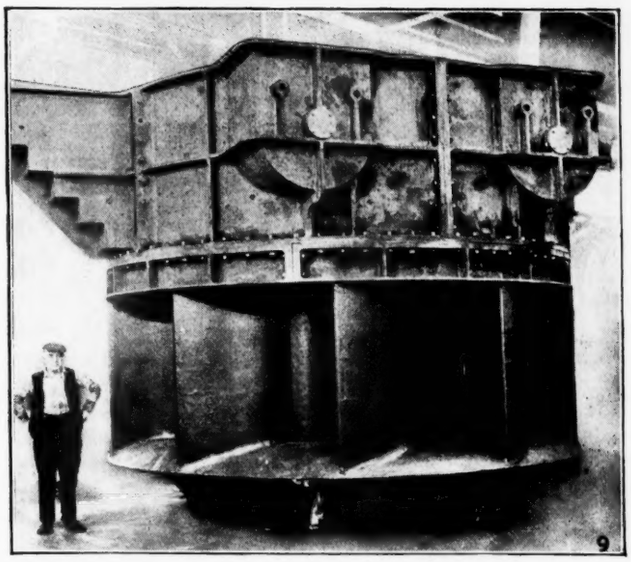
One of these joints is provided at the power house and another at the forebay headworks, with three at intermediate points. On the inside the penstocks have been waterproofed with an iron compound. To maintain the temperature change at a minimum in the concrete, the top surface of the penstocks has been covered with three feet of sand and gravel. Extending from the floor level just outside the downstream side of the gate house a two-foot square air vent and inspection well is provided for each group of three penstocks and is equipped with a steel ladder for easy access to the penstocks for inspection. This vent is placed between the two of the penstocks in the group, and access is had to all three by means of a two-foot vertical slot extending the full inside height in the intermediate walls.

The water velocities throughout the development have been maintained low so as to obtain high over-all efficiency. These velocities are, with full load on the plant through the canal headgates 3.36, in the canal 4, under the curtain wall 2.28, through the racks 3, through the penstocks gates 4 and in the penstocks 5.23 ft. per second.

Flow of Water to Turbines To Be Metered

Provisions have been made to use the pressure differential due to the change in size of the water passage from the penstocks to the scroll case as a means of obtaining both a direct indication and a continuous record of the water passing through the turbines. Four piezometer connections have been made at the lower ends of the penstocks and are connected to a manifold. The same number of connections have been made around the entrance to the scroll case and connected into another manifold. These two manifolds at each penstock will be piped to the switchboard room, where the instruments will be located. Thus the operator will be able at all times to determine the over-all efficiency of each unit and the plant. Provisions are also made for checking the sum of the meter readings by a gaging station in the canal. Calibration of the meters will be done by checking them against the Allen salt-velocity or the Gibson time-pressure method of water measurement, connection having been provided in the penstocks to use either of these methods. Thus it is expected that it will be possible to have a complete record of this plant's performance as in any steam plant.





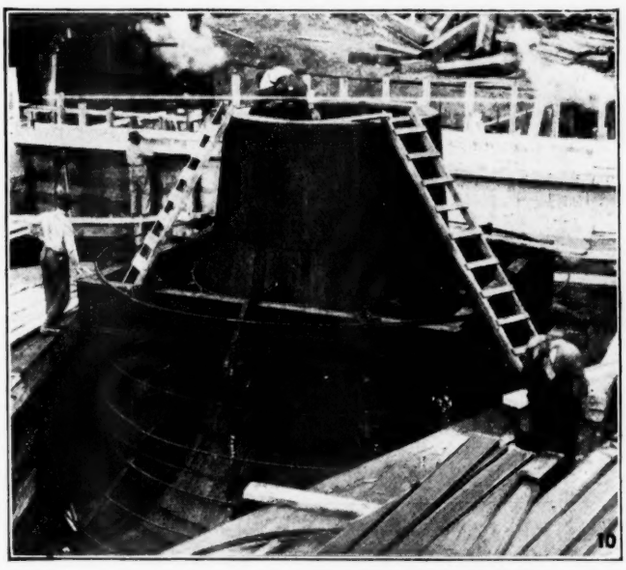


FIG. 8—ONE OF MAIN GENERATORS, RATED AT 9,000 KVA. AT 80 PER CENT POWER FACTOR.

FIG. 9—ASSEMBLY OF ONE OF THE CAST-IRON PIT LINERS AND SPEED RINGS.

FIG. 10—STRUCTURAL-STEEL REINFORCING AND FORM FOR ONE OF THE DRAFT-TUBE BELLS. THE STEEL STRUCTURE WAS SUPPORTED ON 4 "H" COLUMNS FROM THE BOTTOM OF THE DRAFT-TUBE CHAMBER WHICH WERE CUT OFF AND REMOVED AFTER THE TUBES HAD BEEN COMPLETED.

In the construction of the power house the chief problem was to support it on an area underlain with quicksand. This difficulty was solved by dividing the foundation area into ten box sections by driving sheet-steel piling to 25 ft. below subgrade, as indicated in Fig. 12. These sections act as sand columns to transmit the load to a safe depth. To prevent the material from squeezing out in front of the foundation, the reinforced-concrete tailrace apron was extended out about 50 ft. from the power house to where gravel was found in the river bed, and heavy concrete buttresses carried up from this apron to the sides of the power-house foundation. Observations showed that during the construction of the concrete foundation for the power house and the draft tubes a progressive settling took place, but none has occurred since.

The draft tubes are of the Moody spreading type with a special construction for supporting the bell of

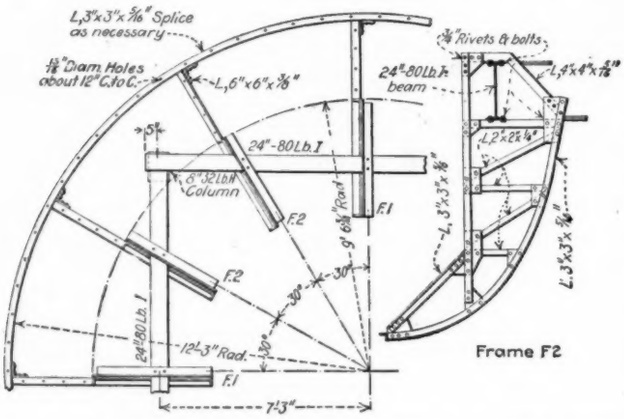


FIG. 11—SECTION OF STRUCTURAL-STEEL FRAME USED FOR REINFORCING DRAFT-TUBE BELLS

the tube. The problem of reinforcing the draft-tube bell and its supporting slab is one of the most difficult in the design of the substructure, on account of the wide span between the supporting piers and the heavy loads that have to be supported. In the construction of these draft tubes twelve structural steel frames, Fig. 11, were hung from a square formed of four 24-in. I-beams. This structure was carried on four 8-in. H-columns, as indicated in Figs. 10, 11 and 13. In addition to this structural-steel vertical reinforcing, the bells have circular reinforcing. After the tubes were completed the four columns were cut off and

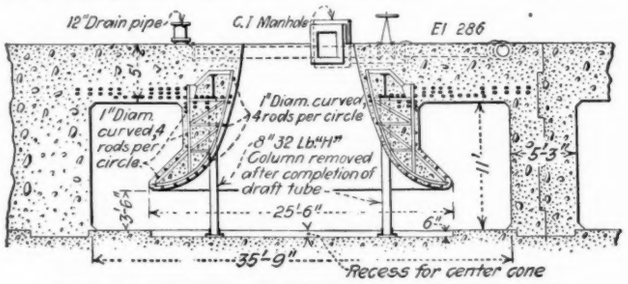


FIG. 13—SECTION THROUGH DRAFT-TUBE BELL

removed. This design greatly simplified not only the problem of reinforcing the tubes, but also their construction. To prevent turbulence in the center of the tubes the cone is extended up to within about five feet of the bottom of the runner.

The scroll cases for the turbines are cast in the concrete substructure. Entrance to the turbine from the scroll case is through a cast-steel speed ring on top of which is bolted a cast-iron pit liner. When pouring the substructure above the scroll case, the wheelpits were left about three feet larger in diameter than required and extended up to within about one inch of the generator floor elevation. After the speed ring and pit liner had been put in place, concrete was poured to bring the wheelpit to the required diameter.

As can be seen from Fig. 14, the units are set off the center line of the penstocks by distances varying from 6 ft. 5 in. to 12 ft. 5 in. This is due to the penstocks being set on 39-ft. 6-in. centers and the turbines on 41-ft. centers. It was originally intended to place the turbines on 39-ft. 6-in. centers so that they would be located the same with reference to the center line of the penstocks. When the excavation for the power house was made, the quality of the material underlying the site was found to be so bad as to warrant raising the bottom of the foundation 5 ft. and increasing the spacing between the units 2 ft. 6 in. This not only

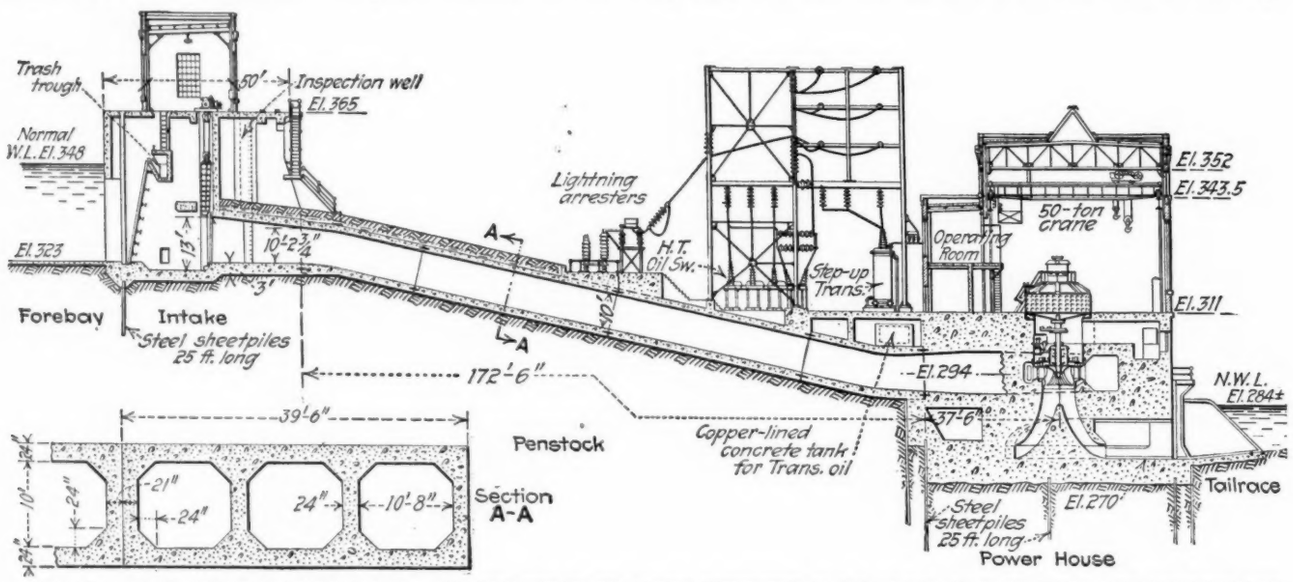


FIG. 12—GATE HOUSE, PENSTOCKS AND POWER HOUSE ARE SUPPORTED ON SAND IN SOME PLACES UNDERLAIN WITH QUICKSAND

simplified the construction, but also reduced the cost. When this change was decided on, construction work on the penstocks and their headworks was well under way, therefore it was necessary to leave their spacing as originally planned and change the location of the turbines to satisfy their new spacing.

Each of the four turbines is of 10,000-hp. capacity and operates at 150 r.p.m. The runners are of the usual mixed-flow or Francis type and have a specific speed of 80 in the foot-pound system or 354 metric. Offset link-type operating gear is used with the breaking links normally in compression and designed to break by shearing if foreign obstructions should lodge between the guide vanes during either opening or closing motions. The operating gear is provided with a grease gun that operates with compressed air so that all bearings can be lubricated by simply turning a cock in the connections between the gun and bearings. A lignum-vitæ guide bearing for the turbine shaft is supported on the cover plate. In addition to this bearing there are two guide bearings in the generator, and the weight of the revolving element is carried on a car-spring thrust bearing supported on a cast-steel framework resting on the generator's stator frame.

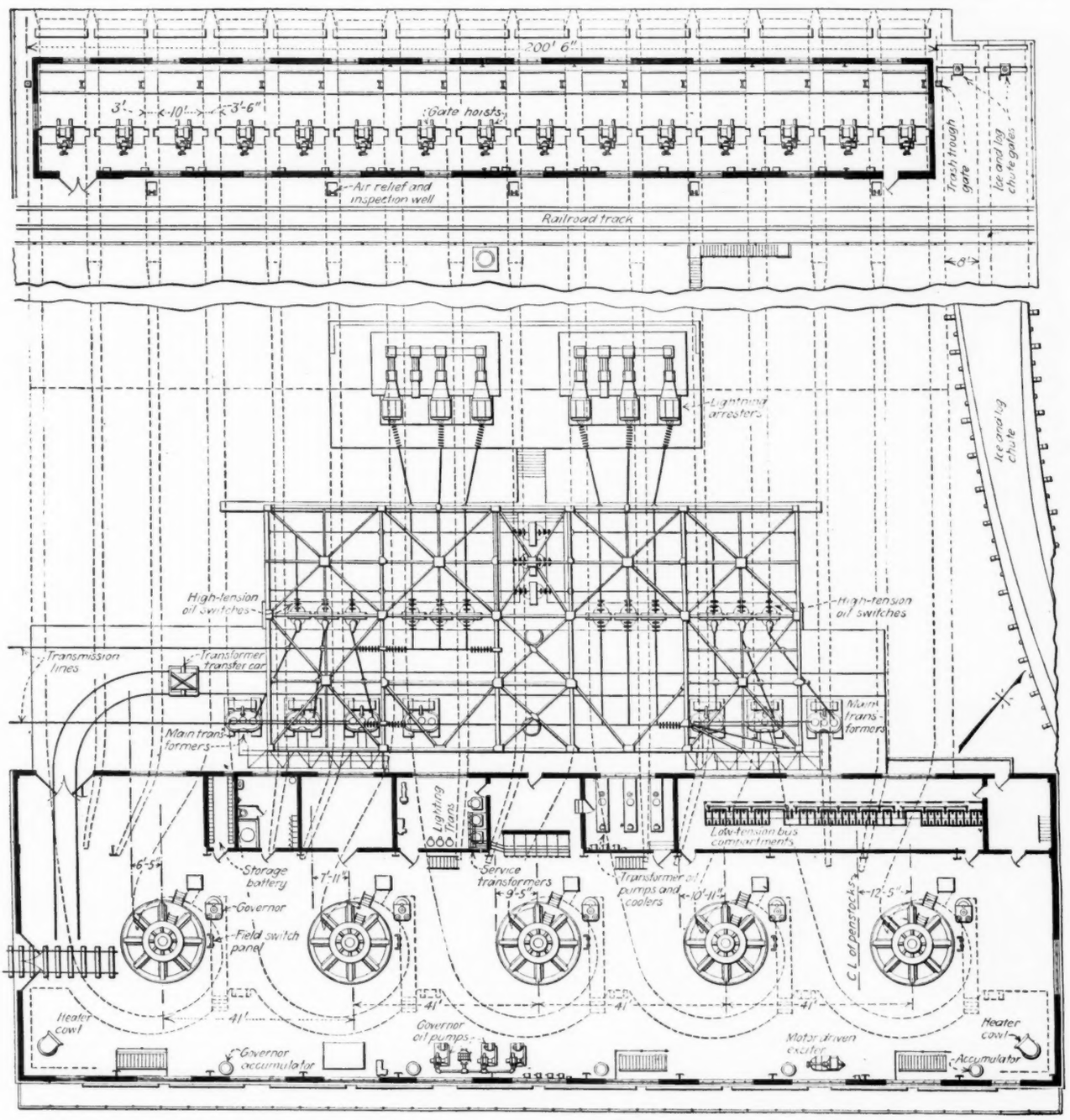


FIG. 14—REINFORCED-CONCRETE PENSTOCKS FORM A SLAB RESTING ON THE SAND BETWEEN THE POWER HOUSE AND HEADWORKS

The main generators are of the standard vertical waterwheel type designed for 6,600 volts, three-phase, 60 cycles at 150 r.p.m. These machines operate in parallel on a 6,600-volt bus. In warm weather ventilating air will be taken from outside the building through openings in the downstream side of the power-

house walls below the generator-room floor. After passing through the machines, the cooling air will be discharged into the generator room and to outside the building through windows in the roof. In the cold weather the ventilating air will be taken from inside the building and recirculated. When this is not sufficient to heat the building, artificial heating will be used. A coal-fired air heater on the turbine-gallery floor is provided at each end of the plant. Circulation of the air through the heaters is maintained by a motor-driven blower.

Excitation is supplied by a direct-connected exciter mounted about the thrust bearing on each machine. Emergency excitation can be supplied to any unit by a motor-driven exciter. Auxiliary power for the station is taken from the 6,600-volt bus through a bank of step-down transformers.

One of the interesting features of the electrical equipment is the step-up transformers. There are seven of these units rated at 7,500 kva. single-phase 6,600 to 66,000 volts. Six of these are connected delta to delta in two banks and to the transmission lines of the Adirondack Power & Light Corporation. It is expected that eventually the transmission voltage will be raised to 114,500, at which time the transformers will be connected star on the high-voltage side. One unit is used as a spare, and provisions have been made for quickly changing transformers in case of failure of one unit in a bank. A railway track runs along in front of the transformer and around into the generator room, so that any transformer may be pushed on to a transfer car and transported to where it can be handled by the station crane. The general arrangement of this track can be seen in Fig. 14. The transformers are on concrete platforms over the penstocks just outside the plant. The oil is water-cooled, and if the cooling coils were located in the top of the transformer tank, as is general practice, there would be danger of the water's freezing in winter, as temperatures of 40 deg. F. below zero are experienced in this section. This difficulty was overcome by pumping the oil from the transformers through cooling coils located in tanks of water inside the power house. The warm oil is taken from the top of the tanks through the cooling coils and then pumped back into the bottom of the transformer tanks.

The general design was made by A. H. White, chief engineer of the International Paper Co., with H. De B. Parsons as consulting engineer. C. S. Colson and W. H. Coventry were in charge of the hydraulic and electrical layout. The details of the structural design were worked out by the firm of Parsons, Klapp, Brinckerhoff & Douglas, of New York, in charge of E. E. Halmos, designing engineer. The construction work was done by the Parklap Construction Corporation under the direction of W. J. Douglas and E. E. Halmos. The field staff consisted of A. Little, manager; W. F. Barnes, superintendent; and G. Miller, supervising field engineer. H. M. Hale was resident for the International Paper Co.

TABLE OF IMPORTANT DATA OF THE SHERMAN ISLAND PLANT OF THE INTERNATIONAL PAPER COMPANY

GENERAL

Location of plant	Sherman Island, N.Y.
Constructed by	Parklap Construction Co.
Present capacity, hp	40,000
Ultimate capacity, hp	50,000
Size of units, hp	10,000
Width of power house, inside	66 ft.
Width of generator room, inside	50 ft.
Width of operating bay	16 ft.
Length of generator room, inside	228 ft.
Distance from generator-room floor to bottom of roof truss, ft	41 ft.

FULL-LOAD WATER VELOCITIES PER SECOND, FOR 5 UNITS

Through head gates into the canal	3.36 ft.
In concrete-lined canal	4 ft.
Under curtain wall, at penstocks	2.28 ft.
Through racks	3 ft.
Through penstock gates	4 ft.
In penstocks	5.32 ft.
Discharge from turbine runner (axial component)	26 ft. 8 in.
Discharge from draft tube to tailrace	4.8 ft.

PENSTOCKS AND GATEHOUSE

Curtain wall extends below normal water level	4 ft.
Racks, two sections high, made of ⅜x4-in. iron on edge, spaced	3½ centers
Trash-Trough Gates:	
Manufacturer	Coffin Valve Co.
Intake valve, size	12 in., circular
Discharge valve, size	18x24 in. cast iron
Head Gates at Penstock Mouth:	
Manufacturer	Champion Eng. Co.
Type	Broome, caterpillar
Size	10x13 ft.
Number per turbine	3
Headgate hoists, electrically controlled from main switchboard	Maine Electric Co.
Two headgates in log- and ice-chute, structural steel covered with concrete	Size 8x12 ft.
Material	Reinforced concrete
Size of each	10 ft. 8 in. x 10-ft.
Provisions made in penstock for measuring the flow by the Gibson Time Pressure Method or Allen Salt Velocity Method and also to obtain a record flow on venturi meters.	

TURBINES, GOVERNORS AND DRAFT TUBES

Manufacture of turbines	Wm. Cramp & Sons Ship & Engine Building Co.
Number installed, present	4
Number installed, ultimately	5
Size, hp	10,000
Flow, cu.ft.-sec. per unit at full load	1,570
Type	Vertical, Francis
Head	63 to 69 ft.
Specific speed, ft.-lb. system at normal head of 66 ft.	80
Horsepower under 1 ft. head	18.7
Speed	150 r.p.m.
Guide bearing, lignum vitae, water lubricated.	
Scroll case, reinforced concrete with cast-iron speed ring, having an inside diameter of 9 ft. 6 in.	
Draft tubes:	
Type	Moody spreading
Bottom of speed ring above bottom of draft tube	19 ft. 9 in.
Bottom of runner above bottom of draft tube	20 ft. 4 in.
Diameter at runner discharge	8 ft. 8 in.
Maximum width of chamber at bottom	35 ft. 9 in.
Width of bell outside at bottom	25 ft. 6 in.
Bottom of bell from bottom of chamber	3 ft. 6 in.
Length of bell from bottom of runner	16 ft. 3 in.
Height of cone	14 ft. 8¼ in.
Size of discharge to tailrace, two openings, 11 ft. high by 15 ft. wide	
Governors:	
Manufacturer	Wm. Cramp & Sons Snip & Engine Building Co.
Type	Double floating lever
Location	Generator floor
Governor pumps; number, 3; capacity, 165 gal. per min. each, type motor-driven rotary.	
Governor fluid; oil, under 185 lb. pressure.	
Manufacturer of governor pumps	Woodward Governor Co
Tachometers	Foxboro Co., Inc.

MAIN GENERATORS

Manufacturer	General Electric Co.
Size, 9,000 kva.; voltage, 6,600; phases, 3; frequency 60 cycles.	
Number of poles, 48, speed, 150 r.p.m.	
Thrust bearing, car-spring type, mounted on top of generator.	
Exciters, direct connected to generators.	

MAIN TRANSFORMERS

Manufacturer	General Electric Co.
Number	7
Type, outdoors; conservator; oil pumped continuously through separate water cooler.	
Size, 7,500-kva	Voltage, 6,600 to 66,000 delta, 114,500 star

MISCELLANEOUS

Storage battery	Electric Storage Battery Co
Transformer oil coolers	Griscom-Russell Co.
Main transformer oil pumps	Buffalo Steam Pump Co.
Water-flow meters	Builders Iron Foundry
Piping and valves	Crane Co.
Valves and fittings	Walworth Mfg. Co.
Traveling crane	Manning, Maxwell & Moore, Inc.
Gate hoist at dam	Manning, Maxwell & Moore, Inc.
Air compressor	General Electric Co.
Heating system	American Blower Co.

All the electrical equipment with the exception of the storage battery for control service was supplied by the General Electric Company.

Effects of Opening the Secondaries of Current Transformers

By Ralph Brown*

Practically every article dealing with the series, or more properly, the current transformer emphasizes the danger arising if the secondary winding becomes open-circuited with current flowing in the primary. The test herein described was made to determine the approximate value of the voltage to be expected across the secondary, with current flowing through the primary of a current transformer having a ratio of 60 to 5 amperes. This transformer, Fig. 1, was designed for use on a 25- or a 60-cycle circuit; the primary consisted of 14 turns of copper strap, with a secondary winding of 164 turns of square copper wire. A small 60-cycle alternator was used to supply the necessary current.

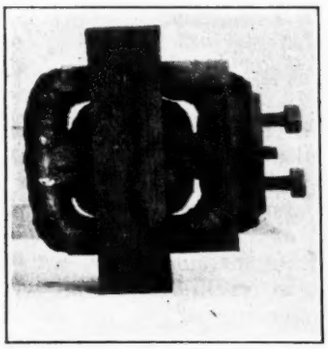


FIG. 1—CURRENT TRANSFORMER ON WHICH TEST WAS MADE

In order that the ampere-turns required to magnetize the iron core and supply the eddy-current and hysteresis losses shall be a small percentage of the total primary ampere-turns and to insure good ratio and phase-displacement characteristics, the full-load primary ampere-turns must be 800 or more. In normal operation, when current is flowing in the secondary, the secondary ampere-turns are nearly equal in number and almost opposite in effect to the primary ampere turns. For this 60-ampere transformer there are 820 secondary and 840 primary ampere-turns. At full load, then, but 20 ampere-turns are available for the core, which is so proportioned that it will be magnetized, at full load, at the low density of about 23,000 lines per square inch, at 60 cycles. The object of working the iron at this low density is to reduce the core loss to a low value.

When the secondary circuit is opened, the secondary current ceases to flow and its demagnetizing effect on the core is zero. The primary current, which under usual conditions is independent of the secondary current, will remain unchanged, with the result that the total number of primary ampere-turns are effective in magnetizing the core. If the primary current is 60 amperes, nearly 840 ampere-turns are magnetizing a magnetic circuit designed for 20 ampere-turns under normal conditions, and a saturated magnetic circuit results. This causes the core losses to be greatly increased above their normal value, and the iron will probably become hot enough, because of these increased losses, to damage the transformer insulation. It is evident that the voltage induced in the secondary winding will be much greater with a magnetizing force of 825 ampere-turns acting on the core, instead of 20, the normal value.

With the transformer connected in series with a water rheostat and a carbon rheostat, one or both being used to vary the primary current in 10-ampere steps, with 200 amperes the effective secondary voltage was, as shown in Fig. 2, 207 volts. The maximum, or crest, voltage was 431 volts, indicating an amplitude factor of 2.08 and proving that the voltage wave was not a sine curve, being distorted therefrom by the presence of harmonics.

When the primary winding was connected directly across the alternator leads, the current being varied by altering the alternator's voltage by means of the field rheostat, 200 amperes induced but 170 effective volts in the open secondary winding. The amplitude factor was not determined under this condition, however; all the tests indicated that the secondary voltage depends not only on the amount of current flowing, but also is greatly influenced by the nature of the load supplied through the primary winding.

An oscillograph, not available during these tests, will be used later to visualize the wave forms of the primary current and secondary voltage and to determine the value of the transient voltage possibly induced at the instant the secondary is open-circuited.

A dynamometer-type voltmeter, the current through which never exceeded 0.05 ampere, was used to obtain the secondary effective voltage, and it was found that the deflections of an electrostatic voltmeter were not influenced by the connection or disconnection of the dynamometer-type instrument. The readings of the crest, or maximum, voltage was taken from the electrostatic voltmeter, the secondary voltage being rectified by a small electron tube.

Since the amplitude factor of a sine wave is 1.41, the value of 2.08 obtained during some of these tests indicates that the maximum value of the voltage induced in

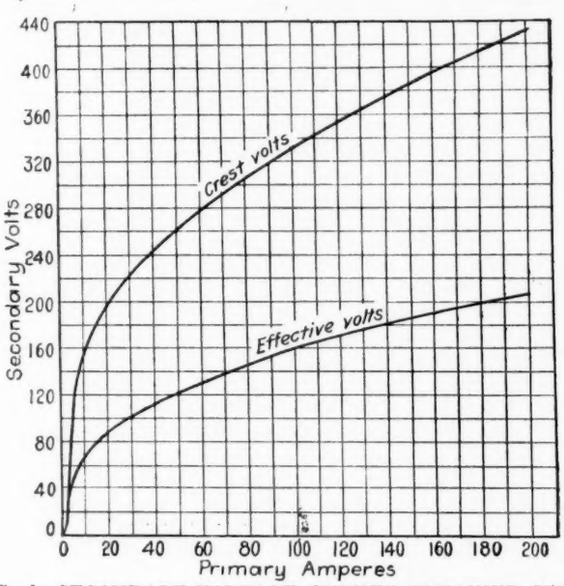


FIG. 2—SECONDARY-VOLTAGE CURVES OBTAINED FROM CURRENT TRANSFORMER

the secondary of a current transformer can be 108, instead of 41 per cent greater than that to be expected from a potential transformer having the same effective voltage induced. In other words, an effective voltage of 100, if it follows a sine curve, as would be the case with a single-phase potential transformer supplying load, would attain a maximum value of 141 volts, while an effective voltage of 100 across the secondary of a current transformer could attain a value of 200 volts or more.

Since it is the maximum value of the voltage that determines the strain on insulation and the severity of an electric shock, it follows that a very severe shock might be obtained from a current transformer, even though a voltmeter would not indicate that a high voltage existed.

*The Great Western Power Company.

Turbine Specifications and Bids for Detroit Municipal Plant

In preparing specifications for power-plant machinery, there is apparently little uniformity in the method of presentation or in the requirements inserted. The tendency is to make them too long and involved, going into too great detail on minor features or calling for something special which results usually in increased cost without commensurate advantage. Often the manufacturer is not given sufficient opportunity to incorporate the results of research in his own particular line and to apply his knowledge to the case in hand. It would seem a better plan to outline briefly the requirements, to prescribe definite limitations with the view of eliminating freak design and to call for duty guarantees under definite conditions. This allows the manufacturer some leeway in meeting the requirements and permits him to bring into play his ingenuity, experience and knowledge often to the advantage of the purchaser. An excellent example of what may be done in this line is afforded by the specifications for the new municipal plant of Detroit, prepared by Smith, Hinchman & Grylls, of that city. The specifications for the turbine-generator units are reproduced herewith. Those for the condenser will appear in an early issue.

THESE specifications and the preceding "General Conditions" and "General Specifications"[1] are intended to provide for all the labor and materials necessary to furnish the turbine-generator units hereinafter specified.

Extent of Work—The following work is not to be included as a part of that covered by these specifications:

(1) Foundations and supporting structures.
(2) External piping connections not a part of the machines.
(3) Condenser with connection to turbine exhaust.
(4) Oil filters and purifiers.
(5) Air ducts, washers and filters for air supply to generator.

All other work, including all labor, materials, equipment and accessories further mentioned or necessary for constructing and installing the turbine-generator units hereinafter specified, shall be done as a part of the work covered by these specifications.

Number Required—Three units or four units shall be furnished at the option of the commission.

Capacity—Each unit shall have a normal rated capacity of 20,000 kw. at 80 per cent power factor.

Description of Unit—Each unit shall consist of a horizontal steam turbine direct-connected to an alternating-current generator with direct-connected exciter.

Turbines—The turbines shall be designed for a normal steam pressure at the throttle of 315 lb. per sq.in. absolute, 200 deg. F. of superheat and an absolute back pressure of 1½ in. of mercury. Under these conditions they shall operate at maximum economy with a steam consumption not exceeding 10.5 lb. per kw.-hr. when the load on each generator is approximately 17,000 kw. at 80 per cent power factor.

Each turbine shall be capable, by governor control only, of driving the generator when the load is 25,000 kw. at unity power factor with steam pressure, superheat and back pressure as stated above.

Each turbine shall be provided with extraction nozzles located, in so far as limitations of design and construction will permit, to supply steam in such quantities and at such pressures as may be required to maintain the heat balance of the plant. The following is an approximation of these requirements:

Nozzle No.	Abs. Pressure, Lb. per Sq.In.	Steam Bled, Lb. per Hr.
1	170	13,000
2	34	10,000
3	4	11,000

All parts of the turbines subject to contact with steam having a temperature above 500 deg. F. shall be of steel or other high-temperature steam-resisting material. The turbines shall be capable of withstanding an initial steam pressure of 365 lb. per sq.in.

[1] These specifications were accompanied by a set of bidding plans showing the general arrangement of the plant and were preceded by a general specification that described the plant contemplated and the conditions under which its equipment was to be built and installed.

SCHEDULE OF TURBINE BIDS

Morrell Street Plant, Public Lighting Commission, City of Detroit

I Bidder			Allis-Chalmers Mfg. Co.		General Electric Co.	General Electric Co.	Westinghouse E. & M.
II Address			Milwaukee, Wis.		Schenectady, N. Y.	Schenectady, N. Y.	East Pittsburgh, Pa.
III Price, 3 units			$939,000	Tolerance	$969,000	$943,500	$945,900
IV Price, 4 units			$1,247,000	Per Cent	$1,292,000	$1,258,000	$1,254,000
1. Delivery time, weeks			80	—5	1st 61 2nd 65 3rd 69 4th 74	1st 61 2nd 65 3rd 69 4th 74	1st 52 2nd 56 3rd 61 4th 65
2. Erection time, weeks			24	—5	8 to 10	8 to 10	8
3. Net weight, turbine, lb			426,000	±3	294,000	282,000	534,000
4. Net weight generator, lb			204,000	±3	185,000	185,000	239,000
5. Net weight complete unit, lb			630,000	±3	525,000	515,000	773,000
6. Steam cons. without excitation, lb. per kw.-hr. with steam at 300 lb. sq. in., 200 deg. F. sup. and 1 in abs. back pressure when gen. output at 0.80 p.f. is	10,000 kw.		10.67	0	11.10	11.15	10.70
	15,000 kw. most economical		10.17	0	10.40	10.50	10.15
	20,000 kw.		10.09 10.27	0 0	10.25 10.40	10.45 10.50	10.05 10.20
7. Max. load steam at 250 lb. sq.in. 150 deg. sup. 1.5 in. abs. back pressure, 0.80 p.f.			20,000	±2	20,000	20,000	19,500
8. Max. blade tip speed, ft. per sec			660		880	880	865
9. Max. stresses, disk, lb. sq.in.			23,500		23,000	24,000	Mean stresses in spindle body 12,000
10. Temp. rise bearings cont. oper. deg. F.			125 deg. F. to 145 deg. F. max.		*to 145.4 deg. based on room at 104 deg. F	*to 145.4 deg. based on room at 104 deg. F	To 130 deg. F. max.
11. Capacity of oiling system, gal			1,400	±5	950	700	1,050
12. Oil circulated, g.p.m.			150	±5	100	170	125
13. Speed reg. 0 to full load, per cent			2	±5	4	4	3
14. Gen. temp. deg. C with 25,000 kva. continuous oper. inlet air 40 deg. C.		stator	100	AIEE	100	100	125
		rotor	125	AIEE	135	135	150
15. Voltage reg., 0 to full load, 0.80			40	±5	30	30	38
16. Exoitation	20,000 kw. at 0.8 p.f.		80	0	100	100	100
Power kw.	25,000 kw. at 1.0 p.f.		62	0	75	75	75
17. Rated capacity exciter, kw					120	120	125
18. Air required, generator, c.f.m.			70,000	0	60,000	60,000	65,000
19. Generator reactance, per cent			16	±10	13	13	13
20. Turbine speed, r.p.m.			1,800		1,800	1,800	1,800

*Temp. rise given as 41.4 deg. F.

absolute and a total temperature of 700 deg. F. without injury to any of their parts.

Generators—The generators shall be three-phase 60-cycle 13,200-volt star-connected machines. They shall be wound for excitation at 240 volts at maximum load. All leads shall be brought to the outside of the machines to permit applying differential protection.

Rotor fans shall be capable within limitations of design of producing sufficient static pressure to circulate the air supplied to the generators through the duct system.

Accessories—Each unit shall be provided with throttle valve, generator and exciter field rheostats, vibration tachometer and generator temperature indicators.

Rheostats shall be equipped with 125-volt direct-current motor-operating devices. Exciter field rheostats shall be of sufficient capacity to reduce the voltage 80 per cent below normal.

One set of wrenches and special tools required by peculiarities of design shall be furnished.

Delivery and Erection—The units shall be delivered and completely erected ready for external piping and electrical connections.

PROPOSAL FORM TO BE COPIED BY BIDDER

(Date) .....192...

(Name of Bidder) .....

Original or Duplicate Proposal for Turbine-Generator Units to the Public Lighting Commission of the City of Detroit.

I. The undersigned proposes and agrees to contract with the Public Lighting Commission of the City of Detroit in the form of contract accompanying the specifications, to furnish all the labor, materials and equipment, and perform all the work called for by the specifications and drawings which they mention, prepared by Smith, Hinchman & Grylls, 800 Marquette Building, Detroit, Mich., entitled: "Specifications for Machinery and Equipment" for Morrell Street Plant of the Public Lighting Commission of the City of Detroit, Section E, "Specifications for Turbine-Generator Units," and agrees to accept as payment therefor the sum of

.....$.....for three units;
.....$.....for four units.

II. The undersigned further proposes that the execution of his work and the character and performance of the completed work will conform to the following, and guarantees that the deviation therefrom will not exceed the tolerances which he has stated:

		Per Cent Tolerance
(1)	Time within which complete shipment will be made after entering into contract, weeks	.....
(2)	Time within which erection will be completed after making shipment, weeks	.....
(3)	Net weight of turbine, lb.	.....
(4)	Net weight of generator, lb.	.....
(5)	Net weight of complete unit, lb.	.....
(6)	Steam consumption of unit without extraction in lb. per kw.-hr. with 300 lb. per sq.in., 200 deg. F. superheat, at throttle and 1 in. absolute back pressure when generator output at 0.8 p.f. is:	
	10,000 kw.	.....
	15,000 kw.	.....
	_____ kw. (most economical)	.....
	20,000 kw.	.....
	(Submit curves based on 0.6, 1.0 and 1.5 in. back pressure)	
(7)	Load that unit will carry when steam at throttle drops to 250 lb. per sq.in., 150 deg. F. superheat and back pressure to 1½ in. absolute, kw. at 0.8 p.f.	.....
(8)	Maximum tip speed of blading, ft. per sec.	.....
(9)	Maximum stresses in disks at normal speed, lb. per sq.in.	.....
(10)	Temperature rise of bearings with unit in continuous operation, deg. F.	.....
(11)	Oil holding capacity of oiling system, gal.	.....
(12)	Oil circulated, gal. per min.	.....
(13)	Speed regulation of turbine, no load to full load, per cent	.....
(14)	Generator temperature (deg. C.) including "conventional allowance" with 25,000 kva. continuous load and 40 deg. C. inlet air:	
	Stator (Method.....)	.....
	Rotor (Method.....)	.....
(15)	Voltage regulation of generator no load to full load at 0.8 p.f., per cent	.....
(16)	Power required for excitation at following loads: kw.	
	20,000 kw. at 0.8 p.f.	.....
	25,000 kw. at 1.0 p.f.	.....
(17)	Rated capacity of exciter, kw.	.....
(18)	Air required by generator, cu.ft. per min.	.....
(19)	Inherent reactance of generators, per cent	.....
(20)	List of spare parts and number which bidder recommends to be carried and price of each at which he will furnish and deliver them with the turbine units:	

Name of Part	*Number*	*Price*
.....	.....	.....

III. If this proposal shall be accepted by the Public Lighting Commission and the undersigned shall fail to contract as aforesaid and to furnish the required bonds within 10 days, not including Sundays, from date of serving written notice from the engineer that the contract is ready for signature, then the undersigned shall be considered to have abandoned the contract, and the certified check accompanying this proposal, for the sum required by the notice to bidders, shall become the property of the Public Lighting Commission. If the undersigned enters into the contract in accordance with this proposal, or if his proposal is rejected, then the accompanying check shall be returned to the undersigned.

IV. Signed and sealed this ..... day of .....192..
(Signature of bidder with business address)

There is no established rule for the size of the receiver on an air compressor. If the character of the work is intermittent, a more generous receiver capacity is advisable. However, it is possible in most cases to store only a relatively small amount of air in a receiver with permissible pressure fluctuation. A rough rule is to provide a storage of approximately one minute's supply of air. Thus for the 2,800-cu.ft. compressor we would have approximately 466 cu.ft. of compressed air per minute, so a cubic content of 400 cu.ft. would be liberal receiver capacity. This would be about 66 in. in diameter by 18 ft. long. The pipe leading the air from the compressor should enter the receiver near the top and leave near the bottom.

Testing an AMMONIA REFRIGERATING PLANT

by Chas. H. Herter

ENGINEERS usually know what the efficiency of a machine should be theoretically, but it requires actual tests to show how close actual results approach the ideal. A refrigerating plant comprises so many elements, all influencing the final result, that tests are expensive and therefore are undertaken but infrequently. On rare occasions the purchaser stipulates that the output of plant is to be measured under specified temperatures, and in such case a test is made to prove whether the guaranteed result is actually obtained. At some other installation other temperatures will prevail, and the output of the same-sized machine will be different, and if no tests at different temperature levels are available, one has to approximate the probable capacity by theoretical reasoning.

Running of Tests

The tests given in the following report show the results obtained with the same equipment while cooling brine at various temperature levels from +8 down to —18 deg. F. The four test runs are purposely numbered in the order of capacity rather than in chronological order, so as to show distinctly how the capacity falls off successively and how the horsepower per ton of refrigerating effect increases continually as the brine and evaporator temperatures are lowered. Since these tests are well adapted for instructive purposes and should be of interest to American engineers, the type of machine used being quite similar to American compressors, they have been recalculated from the metric to the English system of units, condensed and rearranged, the original values being in the metric system as is used with German tests.

The refrigerating machine and brine covering plant were built by A. Borsig, manufacturer of NH_3 and SO_2 ice machines, and erected at the municipal meat-packing plant at Berlin. Directly after installation the equipment was to be subjected to several acceptance tests. The local conditions were particularly favorable for this purpose, because during test the load could be carried by other machines in the slaughter house. Primarily, the new plant was to furnish brine at —4 to —8 deg. F. for the meat-freezing rooms, but to test the machine at a higher and at a lower temperature, a total of four runs were made.

The ammonia compressor is of the single-cylinder, horizontal double-acting type, 18½-in. bore and 27⅝-in. stroke; diameter of piston rod, 3⅛ in.; average revolutions per minute, 84.2, corresponding to a mean piston speed of 387 ft. per minute. The suction pipe was 5.9 in. in diameter (6.3 in. outside diameter), and 98 ft. long; discharge pipe, 4.92 in. in diameter (5.24 in. outside diameter), length 72 ft.; liquid line, 2.03 in. in diameter. The inside areas of these pipes are one-third larger than areas used in the United States. The compressor has a water jacket and is belt driven by a 250-hp. synchronous motor.

The condenser battery consists of two groups of five stands each, one group being sufficient in the case of cold water or light load. Each stand includes 19 double pipes, the inner or water tube being 1.495x1.662 in. in diameter, the outer or gas tube being 2.50x2.757 in. in diameter, giving a total water-cooled surface of 1,260 sq.ft.

The Evaporating System

The brine tank is 16.4x11.8 ft., 8.2 ft. high, with a longitudinal partition in the center. Even with this the brine velocity due to the wide tank is slower than it should be, but local conditions prevented the use of a narrower tank. The steel brine-cooling coils are 20 in number, the pipes being 1.18x1.496 in. in diameter, with a total exterior surface of 3,230 sq.ft. The coils are arranged in three sets, with room left for a fourth group. The coils are operated on the Schmirz flooded system, which means that a large vertical cylindrical liquid separator or accumulator, with an outer surface of 42 sq.ft., is mounted 14 ft. 5 in. above the top edge of the brine tank. The suction vapor from the outlet manifold of the coils enters the side of the separator about one-third from the bottom. Here also enters the fresh liquid coming from the expansion valve. The two mix in the separator, all liquid becomes cooled to the evaporator temperature and passes through the downtake to the inlet manifold of the coils, while the return gas and the 17 per cent of vapor formed in the precooling of the fresh liquid combine and leave near the top of the separator and flow through the suction pipe to the compressor. Where it is not convenient to elevate the separator to a height sufficient to overcome the resistance to flow in the various coils, the precooled liquid is forced by means of an ammonia pump into the line leading to the inlet manifold. With sufficiently low velocity only dry saturated vapor will leave the separator and thus the machine operates as a dry compressor, the efficiency of which is materially better than

in a wet compressor because the dry gas is less influenced by the hot cylinder walls.

The brine tank is insulated all over with 8-in. corkboard, this corkboard being adulterated with wood chips. The separator and suction line are insulated with 2-in. cork covering.

Agitation is effected by two brine propellers about three feet above the tank bottom, the brine passing through both in series.

To maintain conditions constant during the tests and to secure a high degree of accuracy, it was decided to create an artificial cooling load by heating the brine with two steam coils, and to weigh the amount of steam condensed. Before the steam was admitted to the coils, it was passed through a large separator and there its pressure reduced from the initial pressure of 85 lb. to 15 lb. gage. This throttling of the steam caused it to be superheated and therefore free from moisture, and thus its heat content was easily ascertained. The condensate was discharged into a tank placed on a scale, the time required for catching a known weight of condensate being noted.

The temperature of the condensate collected from both steam coils was taken every five minutes. Several brine thermometers showed momentary differences up to 1 deg. F., but kept in step throughout the tests, hence only two central thermometers had to be used to obtain the true mean brine temperature. Indicator diagrams of the compressor were taken every 20 minutes and the r.p.m. taken by a speed counter. Fig. 1 shows a pair of typical diagrams taken during test 3. As reproduced 3 in. long, the spring scale is 100 lb. per inch of height. The clearance volume is about 1.5 per cent. The diagrams show an apparent volumetric efficiency averaging 86 per cent. Presumably Gutermuth spirally wound plate valves are used.

Condenser Records

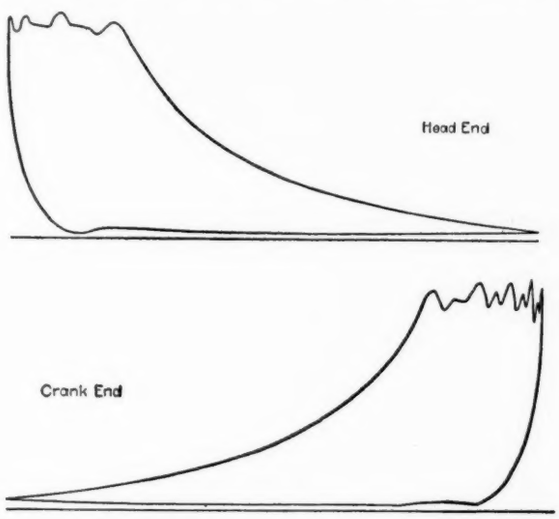
The inlet and outlet temperature of the condenser water was taken every five minutes with mercury-in-glass thermometers graduated to $\frac{1}{10}$ deg. C. The temperature of the jacket water was taken to be identical with that of the condenser-water supply, but its outlet temperature was measured separately. At intervals the rate of flow of the jacket water was ascertained by noting the time required to fill a four-gallon bucket. The water supplied to the ammonia condenser was passed through a Woltmann meter as made by Siemens Halske.

From the table it is apparent where the thermometers and pressure gages were placed, and for identification they are shown numbered. It is to be noted that the discharge pressure gage at the board p_3 did not indicate correctly. Also, the two thermometers t_1 and t_2 evidently were located too close to the expansion valve and therefore were influenced by conduction due to the quick change in temperature at this point. Thus t_1 indicated low and t_2 indicated high. The consumption of current was measured by wattmeter.

At all tests the conditions remained constant and in equilibrium, consequently a test period of two hours was quite sufficient. As the brine temperature was kept practically constant during each test, its volume does not affect the results. The heat imparted to the cold brine by the submerged steam coils has to be absorbed by the refrigerating machine. A small flexible copper tube was usually connected with the steam coil to permit raising the coil out of the brine as soon as it was noticed that the brine tends to rise above the temperature desired.

The horsepower, item 42, is based on 33,000 ft.-lb. per min. The tonnage, item 43, is based on removing heat at the rate of 200 B.t.u. per minute (288,000 B.t.u. per 24 hours). As the brine temperature, item 2, is lowered, the compressor horsepower, item 42, decreases also, but item 43 shows that the horsepower per ton increases. The latter is accounted for by the greater volume required per pound (item 59), by the greater re-expansion in the clearance space as the ratio of compression is increased (item 62) by the smaller refrigerating effect per pound (item 58), and by the higher temperature of the cylinder walls (item 63), which causes the gas to expand during the suction stroke.

For the benefit of those who are interested in comparing practice with theory, comparisons are added



COMPRESSOR DIAGRAMS TAKEN DURING THE TESTS

with an ideal machine operating on the Carnot cycle. To arrive at the practical volumetric pumping efficiency of the compressor, the volume per pound of the superheated vapor at t_5 and p_5 must be used and multiplied by the actual number of pounds of vapor circulated. In addition the machine is to be credited with the cooling equivalent of the superheat acquired by the vapor outside of the compressor. To save space, these refinements are here omitted.

Computed Efficiencies

The efficiencies given differ slightly from those in the original report, because in the present article the properties of ammonia were taken from the new tables issued by the Bureau of Standards, Washington, D. C.

The discrepancies, item 55, in the heat balance can be reduced by taking into account the heat entering the system through the insulated surfaces of the brine tank, separator and its piping. Owing to lack of space these calculations and assumptions are omitted. Even as it is, the practical results check within about 1 per cent, but in comparing the indicated horsepower required per ton of refrigeration with other tests, it should be borne in mind that in the present case the output of the compressor, item 37, is to be increased by the radiation losses from all insulated surfaces, this amount varying from, say, 0.5 per cent in Test 1 to 1.5 per cent in Test 4. When this correction is made, the indicated horsepower per ton under like temperatures

OBSERVATIONS AND DEDUCTIONS FROM TESTS MADE OF A 125-TON AMMONIA REFRIGERATING PLANT

Item		1	2	3	4
	No. of test	1	2	3	4
	Date, 1919	Oct. 7	Oct. 3	Sept. 30	Sept. 27
	Time of day	3:15 to 5:15	11:20 to 1:30	12:30 to 2:30	9:32 to 10:39
	Test period, minutes	120	130	120	67
1	Rev. per min. of compressor	84	84	84.3	84.5
	Temperatures, Deg. F.:				
2	Brine (constant temperature), t_b	+8.6	+0.23	—5.8	—17.95
3	Liquid ammonia before expansion valve, t_1	59.36	53.96	54.14	59 (?)
4	Liquid ammonia after expansion valve, t_2	+0.50	—8.	—5.08	—17.68
5	Liquid from separator, precooled, t_3	—5.62	—11.2	—15.52	—25.6
6	Suction gage temp. at board, t_4	0.68	—4.9	—9.76	...
7	Suction gas thermo. at compressor, t_5	—4	+25.	+23.54	+30.2
8	Discharge gas at compressor, t_6	179.6	202.	203.	224.6
9	Discharge gas at condenser, t_7	177.8	198.14	195.8	204.8
10	Saturation temp. at condenser, t_8	66.2	64.4	62.6	65.3
11	Saturation temp. at board (faulty), t_9	69.44 (?)	68.18 (?)	67.46 (?)	...
12	Liquid from condenser, t_{10}	59.9	55.4	56.12	58.37
13	Condenser water on, t_{11}	56.08	53.83	53.64	54.5
14	Condenser water off, t_{12}	69.66	66.45	64.6	67.95
15	Water leaving jacket, t_{13}	71.6	71.42	70.7	76.64
16	Room temperature, t_{14}	71.6	68.	68.	67.1
17	Actual temp. of steam after reduc. valve, t_{15}	314.6	305.92	290.12	293.9
18	Saturation temp. of steam after reduc. valve, t_{16}	309.2	289.4	244.94	271.25
19	Mean temp. of steam condensate, t_{17}	87	73.4	59.43	152.6
	Pressures Abs., Lb./Sq.In.:				
20	Suction pressure at board, lb. per sq.in. P_4	30.9	26.99	24.2	...
21	Suction pressure at compressor, P_5	27.59	24.11	21.99	14.96
22	Discharge pressure at compressor, P_6	126.5	120.42	117.3	121.18
23	Discharge pressure at condenser, P_8	120.2	116.54	112.8	118.46
24	Discharge pressure at board (faulty), P_9	127.5	124.7	123.11	...
25	Steam pressure after reducing valve, P_{16}	76.8	57.35	27.72	42.7
	Quantities:				
26	Condenser water, gal. per min.	298	265.7	260	141.5
27	Jacket water, gal. per min.	4.8	4.04	3.82	4.04
28	Total water consumed, gal. per min.	302.8	269.74	263.82	145.54
29	Compressor input at switchboard, kw.-hr	144.8	128.6	122.5	108.6
30	Steam condensed, lb. per min.	24.45	20.25	15.91	11.18
	Results:				
31	Total heat of saturated steam, B.t.u. per lb.	1,181.6	1,176.14	1,162.4	1,170.58
32	Amount of steam superheat, B.t.u. per lb. (item 17—item 18) 0.54	2.92	8.6	17.2	11.33
33	Heat content of superheated steam, B.t.u. per lb.	1,184.52	1,184.74	1,179.6	1,181.91
34	Deduct heat content of condensate above 32 deg. F. (item 19-32)	55	41.4	27.43	120.6
35	Net heat absorbed by steam, B.t.u. per lb.	1,129.52	1,143.34	1,152.17	1,061.31
36	Net brine cooling per min., B.t.u. (item 30×item 35)	27,610	23,130	18,330	11,850
37	Dividing item 36 by 200 gives tons refrigeration	138.05	115.65	91.65	59.25
38	Mean effective pressure, crank end, lb. per sq.in.	46.1	46.4	42.8	...
39	Mean effective pressure, head end, lb. per sq.in.	49.6	46.1	42.23	...
40	Indicated hp., crank end (comp. constant = 0.0179)	73.75	69.7	64.5	...
41	Indicated hp., head end (comp. constant = 0.01873)	78.25	72.5	66.7	...
42	Indicated hp., total	152	142.2	131.2	115
43	Indicated hp. per ton ref. effect (item 42 ÷ item 37)	1.102	1.23	1.432	1.94
44	Coefficient of performance (ref. effect ÷ input)	4.285	3.84	3.3	2.43
	Heat Balance				
45	Heat abstracted from brine, B.t.u. per min.	27,610	23,130	18,330	11,850
46	Heat equivalent of compressor work, B.t.u. per min. (item 42 × 42.4)	6,450	6,030	5,560	4,880
47	Heat equivalent of brine agitators (3 hp.), B.t.u.	130	130	130	130
48	Heat supplied, total, B.t.u. per min.	34,190	29,290	24,020	16,860
49	Temp. rise of condenser water, deg. F.	13.58	12.62	10.96	13.45
50	Temp. rise of jacket water, deg. F.	15.52	17.59	17.06	22.14
51	Heat removed by cond. water, B.t.u. per min. = gal. min. × 8.33 × deg. temp. rise.	33,700	27,940	23,730	15,870
52	Heat removed by jacket water, B.t.u. per min. = gal. min. × 8.33 × its temp. rise.	620	592	542	745
53	Heat removed by cooling water, B.t.u. per min.	34,320	28,532	24,272	16,615
54	Discrepancy relative to item 48, B.t.u. per min.	+130	—758	+252	—245
55	Discrepancy relative to item 48, per cent	+0.38	—2.59	+1.05	—1.45
	Efficiencies				
	Comparison with a perfect ammonia compressor working with the same evaporating and condensing pressures:				
56	Heat content H of vapor at t_3	609.9	608	606.55	602.34
57	Less heat content h of liquid at t_{10}	109.1	104	104.83	107.34
58	Net refrig. effect in B.t.u. per lb.	500.8	504	501.72	495.00
59	Specific volume V of vapor aspirated at t_3, cu.ft. per lb.	10.38	11.836	13.137	17.724
60	Total vapor aspirated, cu.ft. per min. $\left(\frac{\text{item } 36 \times \text{item } 59}{\text{item } 58}\right)$	572.5	544	480.5	424
61	Displacement of compressor (piston rod deducted) 8.39 × r.p.m. = cu.ft. per min.	705	705	707	708
62	True volumetric efficiency $\left(\frac{\text{item } 60 \times 100}{\text{item } 61}\right)$ per cent	81.25	77.2	68.0	59.8
63	Temperature of vapor at end of adiabatic compression to condenser pressure, deg. F.	209	214	218	260
64	Heat content of this vapor at P_8, B.t.u. per lb.	717	719	724	747
65	Work of compression (item 64—item 56), B.t.u. per lb.	107.1	111	117.5	145
66	Performance E_p of perfect compressor $\left(\frac{\text{item } 58}{\text{item } 65}\right)$	4.68	4.54	4.27	3.415
67	Performance E_a of actual compressor (item 44)	4.285	3.84	3.3	2.43
68	Ratio $E_a \div E_p$	0.916	0.846	0.773	0.712
	Performance E_t of an ideal refrigerating machine operating between the temperature of the brine and the mean temperature of the cooling water is as follows:				
69	Mean temperature of cooling water $\left(\text{item } 13 + \frac{\text{item } 53}{\text{item } 28 \times 8.33} \times 0.5\right)$ deg. F.	62.88	60.18	59.16	61.35
70	Performance $E_t \left(\frac{\text{absolute brine temp.}}{\text{temp. diff.}}\right) = \left(\frac{460 + \text{item } 2}{\text{item } 69}\right)$	8.64	7.68	7.175	5.58
71	Ratio $Ea \div E_t \left(\frac{\text{item } 67}{\text{item } 70}\right)$	0.4965	0.500	0.460	0.436
	Coefficient of Heat Transfer K of Brine Cooling Coils. (Flooded System)				
72	Heat absorbed by coils $\left(\frac{\text{item } 45 \times 60}{3{,}230 \text{ sq.ft.}}\right)$, B.t.u. sq.ft. per hour	513	430	2,340.3	220.2
73	Temperature difference between t_b and t_3, deg. F.	14.22	11.43	9.72	7.65
74	Over-all heat transfer coefficient K, B.t.u. per hr. per sq.ft. per deg.	36.08	37.64	35.0	28.8
75	Velocity V_a of saturated vapor through 5.9 in. pipe, (item 60 ÷ 0.1898 sq.ft.), ft. per min	3,020	2,868	2,533	2,235
76	$\sqrt{\text{Vapor velocity}} = \sqrt{v_a} =$	54.94	53.55	50.33	47.25
77	Herter's approximation for K: $c\sqrt{v_a}$ yields in this case $\frac{3}{5}\sqrt{v_a}$ = B.t.u. per hr. per sq.ft. per deg.	36.6	35.7	33.6	31.5

agrees with the values found with American double-acting compressors. It will be noted that in Tests 2, 3 and 4 the suction gas reached the compressor superheated by 30 to 45 deg. the difference between t_1 and t_s. If this had not been the case, the power per ton would have been lower.

The efficiency of power conversion from switchboard to compressor piston may be obtained from items 29 and 42. Thus, in Test 1, 144.8 kw.-hr. $\times$ 1.36 = 197, and 152 $\div$ 197 = 0.771, showing a total loss of 23 per cent, of which the loss in the motor may be 10 per cent. in the belt drive 3 per cent, and in the compressor 10 per cent. With a direct-connected motor a saving of 3 to 5 out of the 23 per cent is obtainable.

It is interesting to deduce from these tests the value of K, the coefficient heat transfer from brine to evaporating ammonia. Line 74 shows how K declines as the load on the cooling surface is decreased. Evidently, it is inaccurate to assume that K in B.t.u. per degree difference remains constant. Throughout the tests the rate of brine agitation in cubic feet per minute remained unaltered, the only difference at the brine side of the coils being a lower temperature and with it a slight thickening of the brine, the influence of which on K becomes noticeable only in Test 4, where the brine was at —18 deg. F. The cause of the variation in the value of K must therefore exist within the cooling coils. Within these coils the rate of evaporation or rate of vapor production falls off as the temperature difference is decreased, the amount of external surface of the coils also remaining constant. Not having sufficient data to determine the resulting vapor velocity in the coils, the writer simply assumed that the variation in coil velocity was similar to that existing in the suction main, although about 17 per cent of the total vapor never passes through the coils, but returnes directly from the liquid separator to the compressor. Line 75 serves to show the falling off in velocity and discloses a relation to exist between V_a and K. This relation is fairly well expressed by writing in accordance with line 77:

$$K = \sqrt[3]{V_a}$$

This formula may be checked by applying it to the example of a modern American can-ice tank where 240 ft. of 1¼-in. standard-weight ammonia pipe is used per ton of ice made per 24 hours in conjunction with a properly functioning flooded system of precooling and circulating the ammonia, and using agitation giving a lively flow of brine over the coils. Assuming one coil 720 ft. long (its outer surface 720 $\div$ 2.301 = 312.8 sq.ft.) a brine temperature of 15.4 deg. F. an evaporating pressure of ammonia in coil, 22.5 lb. = gas at 8.5 deg., making the temperature difference, 15.4 — 8.5 = 6.9 deg.; output of coil, 3 tons ice, which at 1.5 tons = 4.5 tons refrigeration is equivalent to a heat removal of 54,000 B.t.u. per hour. Hence in this case,

$$K = \frac{54{,}000}{312.8 \times 6.9} = 25 \text{ } \textit{B.t.u. per sq.ft. per deg. F. difference;}$$

The exit velocity from this coil would be

$$V_a = \frac{4.5 \textit{ (tons)} \times 0.43 \textit{ (lb.)} \times 7.55 \textit{ (cu.ft)}}{0.01039 \text{ (sq.ft.)}} = 1{,}406 \textit{ ft. per min.,}$$

and so we should expect to obtain in this case a value of

$$K = \sqrt[3]{1{,}406} = 25 \textit{ B.t.u.}$$

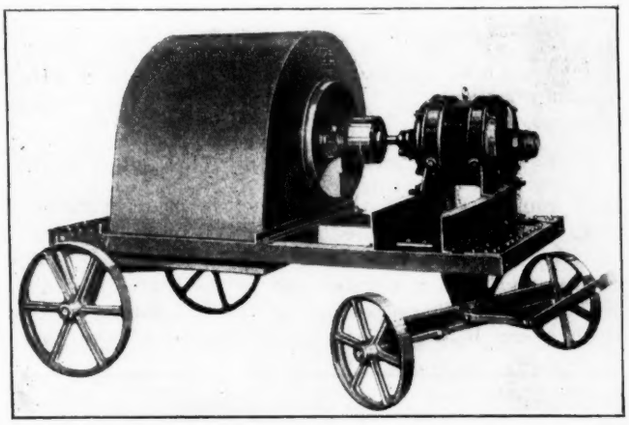
In an effort to secure a high value of K and high velocity V_a, one may be tempted to use long coils per feed valve, but the gain resulting may be nullified by the greater pressure and temperature drop within a long coil, as pointed out in an article by the writer in *Power*, July 24, 1923. However, it is proper to deduct from coils the vapor resulting from precooling in the accumulator.

It may be added that values of K as high as those given on line 74 in the table have scarcely ever been reported from American-made submerged brine coils, and until further proof is produced, it is advisable to base the quantity of piping on $K = 0.6\sqrt{V_a}$ rather than on $0.66\sqrt{V_a}$.

The values of K in the double-pipe condenser cannot be calculated because the various surfaces used for removing superheat, for liquefying and for subcooling of liquid are not known.

Portable Fan Equipment for Furnace and Tunnel Cooling

Portable motor-driven fan units have been added to the line of fans manufactured by the Buffalo Forge Co., Buffalo, N. Y. These portable units are especially adapted to furnace cooling in steel mills and ventilating tunnels or other services where a supply of air is required temporarily and where the installation of a permanent unit is not warranted.



PORTABLE MOTOR-DRIVEN FAN UNIT

The illustration shows a typical fan unit of 15,000 cu.ft. capacity against 2½-in. static pressure at 1,150 r.p.m. The fan and motor are mounted on a four-wheel truck with a handle attached for hauling purposes. The unit can be handled easily from one part of a building to another by two men. The fan is of the duplex conoidal type and is direct connected to the motor by means of a flexible coupling.

When an absorption refrigerating plant is running at full capacity, the speed of the ammonia pump should be such that from 0.5 to 0.6 gal. of aqua ammonia will be handled per minute per ton of refrigeration per 24 hours. If in the case of a steam pump, it does not run steadily, but with a jerk, it is probable that there is some difficulty with the valves. The pump piston rods should be kept smooth and true. Any difficulty experienced in properly packing the pump can usually be traced to the fact that the rod has worn tapering. A properly packed box requires only a slight pressure of the gland and follower on the packing.

Fedco Protectometer Systems for Stored Coal

As a means of detecting any case of heat development in sufficient time to permit the removal of hot coal before any serious loss of heat value has been incurred or before actual fire breaks out, a system developed by the Protectometer Company, of Jersey City, N. J., will be of interest to all those who store coal in considerable quantity. The instruments comprising this system are of two types—sentinel instruments, to give warning when the maximum safe temperature has been passed, and dial instruments, for more accurately plotting hot spots and determining the exact location and temperature of coal in the danger zone.

The calibrated dial instrument, used to locate more accurately and plot the hot spots and indicate temperatures at different points in the coal pile, is shown at the right of Fig. 1.

Fig. 2 shows a portion of a coal pile with two sentinels installed. These sentinels are placed permanently in the coal as soon after storing as possible, and on centers approximately 15 to 20 ft. apart. These sentinels are left in this position until the signal indicates excessive heating or until the coal is removed. Each instrument has a sensitive range to guard approximately 150 to 300 tons of coal, piled 15 to 30 ft. high.

FIG. 2—SENTINELS IN COAL PILE, DISCHARGED SENTINEL AT RIGHT

The construction of the sentinel is clearly shown in Fig. 3. It consists essentially of an outside casing *A* of ¾-in. iron pipe treated to resist corrosion and fitted at one end with a pointed cap to facilitate pushing it into the pile. The other end of the instrument is fitted with a cap held in place under tension of the spring *B*, by a series of lengths which are made of a special alloy and chemical compound fusible at a temperature of 150 deg. F. The principle involved is that the development of this temperature in any part of the pile causes the nearest link to fuse which, in turn, releases a signal head that is thrown upward by pressure of the spring *B*. These signals are visible for several hundred feet, and a casual inspection will thus reveal the danger point at which the coal is heating. A discharged sentinel is shown at the right of Fig. 2.

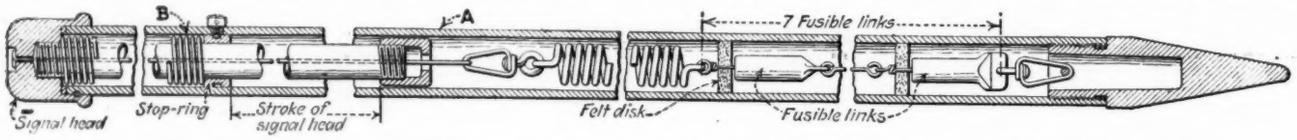


FIG. 3—CONSTRUCTION OF SENTINEL. LINKS FUSE WHEN TEMPERATURE OF COAL REACHES 150 DEG. F.

The fusible links are made in two grades, those that fuse at 125 deg. F. and those that fuse at 150 deg. The 150-deg. link is used for most installations, the 125-deg. link being made for those who want an earlier warning.

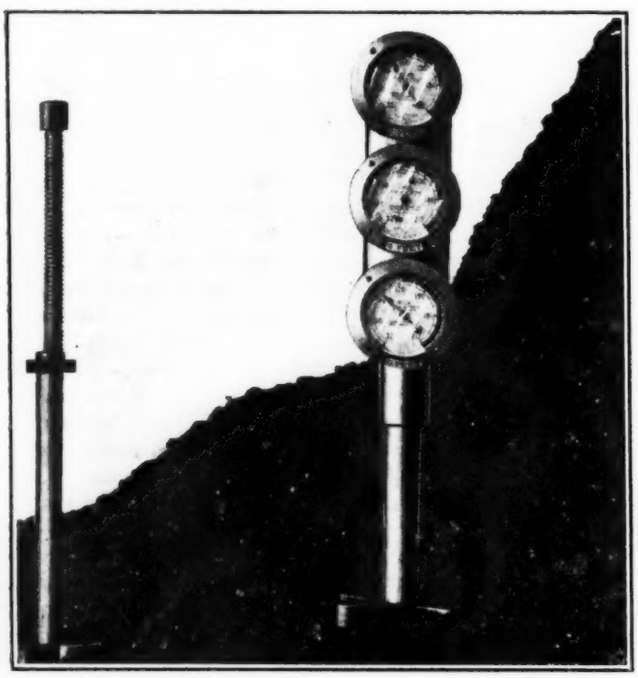


FIG. 1—INSTRUMENT FOR LOCATING THE HOT SPOTS IN THE COAL PILE

There is no need for laboratory precision in the apparatus measuring the coal and water, according to an article on "Boiler House Records" in a recent issue of *Engineering* (London). If a commercial accuracy of 2 per cent is obtained, it is all that is necessary, for it is doubtful whether the integrating wattmeters which measure the electrical output of the station can be relied upon within this limit of accuracy. It is the continuous registration of performance, shift by shift, so that comparisons can be made, which is the essential matter, and not the perfect accuracy of individual observations. Elaborate boiler tests, conducted at intervals, are of little more than academic interest. They demoralize the station while they are being prepared for and carried out, and do much less to promote efficiency of operation than simple continuous records. Even if they indicated what a boiler could do under working conditions, they certainly do not show what it does do, for within half an hour after an efficiency of, say, 80 per cent has been shown on test, the boiler may be operating at an efficiency of 60 per cent or even less without anything being obviously wrong.

America, foremost in business, has a per capita fire loss of $4.75. England's is 72 cents.

What Is the Value of the Hydrostatic Test?

By THOMAS H. FENNER
Chief Engineer, General Accident Assurance Company of Canada

THE question of applying a hydrostatic test to pressure-containing vessels, and particularly to steam-boilers, is one that frequently arises. It is a subject on which there is considerable difference of opinion, as between operating engineers, manufacturers, and those whose duty it is to pass upon the fitness of the vessels for their appointed service. It is the object of this article to show where the hydrostatic test can be of great assistance in determining the condition of the object under examination, and also other cases where it may be a detriment to the safety which it seeks to insure.

IT MAY safely be said that if hydrostatic tests were carried out in every case with all the necessary precaution and careful taking of measurements, no danger would be involved and each test would be efficient. Unfortunately, in the great majority of cases where the test is applied on boilers that have been in use for any length of time, none of these precautions is observed. On this account the last state of these boilers is very often worse than the first.

Reduced to its elementary principle, the application of a hydrostatic test may be said to be the subjecting of the object tested to something more than its working pressure, in an effort to discover if this object comes through without signs of weakness. It is equivalent to testing a bridge by putting on it a load greater than it will ever be expected to bear in ordinary practice, the deduction being that if it comes through safely, there will be no danger of its failing under the working load of every-day practice. However, in both the bridge and the boiler the safe working load has been determined by careful calculation, and the hydrostatic test on the boiler and the load test on the bridge bear a definite relation to the working load. The tests, as carried out in new construction, are intended to discover any local weakness, such as defective workmanship, that may develop under stress. If the stress is greater than that to be encountered under ordinary conditions, and no such weaknesses develop, it is proof that the construction and workmanship are sound. The hydrostatic test on new boilers almost invariably discloses small defects such as leakage at the seams and rivets, which are remedied by calking. It is the usual practice to make the test pressure on new boilers double that of the safe working pressure as determined by calculation. The pressure is applied gradually, and the small defects that appear as the pressure is increased are followed up and remedied before going further. If pressure is applied suddenly, it is difficult to locate accurately the number of small leaks that are liable to appear, and the job is made that much harder and the time taken longer in proportion.

In the ordinary type of horizontal fire-tube boiler there is little possibility of deformation of a new boiler while undergoing a test. In boilers of the internally fired type, such as the vertical fire-tube and the Scotch marine, and locomotive type, where portions of the boiler are under compressive stress, local deformations may appear. The shell of the boiler in either type will of course undergo a slight increase in diameter. These deformations will disappear with the removal of the pressure, but it is possible, when the boiler is on the floor of the shop, to measure the various parts accurately, before, during and after the test, to determine the amount of deformation under the stress and also if the parts resume their original dimensions when the test pressure is removed. Mr. Stromeyer, an eminent English authority on boiler construction and operation, states that the increase in the length of a boiler shell while under test is about one-thousandth of the circumference.

The method of making the measurement of the circumferential stretch is to coil a piano wire one and a half times round the shell and hang a weight on each end. If a mark is made on the two parts of the wire on top of the boiler before the test is made, then, when the test is on, the distance these two marks have moved can be measured. After the pressure is removed, they should again coincide. Mr. Stromeyer states that in most cases they do not exactly coincide, neither is the stretch quite as much as one-thousandth of the circumference, but he gives the friction of the wire on the shell as the reason for this discrepancy. Measurements of furnaces and combustion chambers subject to compressive stress can be made by trammels or telescopic gages. It is possible by means of these measurements, the tensile strength of the materials and the yield point being known, to calculate accurately the stress actually carried by each part during the test. It is not necessary to go into that here, and it is outside the scope of the operating engineer, for whom this article is intended. These particulars are given to impress the fact that stress is always accompanied by strain, and undue stress may result in a permanent strain or deformation of the material.

While it is not the usual thing for a boiler manufacturer to make the measurements given herein, there is sufficient experience of tests made under these conditions, to show what generally takes place. With boilers constructed of material, every chemical and physical characteristic of which is known and in which rigid inspection has been made at every step of the manufacture, such refinement in testing would be a waste of time and money. The ordinary hydraulic test, to double the working pressure, carried out carefully and followed up step by step by remedying such defects as do appear, provides a reasonably safe guarantee to the purchaser and the public that all is well with the boiler before it leaves the manufacturer's hands.

It may be said then, that such a test has a definitely useful object, and achieves it as far as possible.

Before going on to the other aspects of hydraulic testing, it is perhaps opportune to give a few brief definitions of the terms used and some few figures on the subject. Thus, when we speak of a stress, we mean a force applied to an object which tends to alter its original shape. If the force is applied in a manner tending to increase the length of the object, it is called a tensile stress. If the force seeks to reduce its length, as in the case of a column supporting a weight, it is a compressive stress. If the stress is such as to cause the material to break by sliding one part over another, it is a shearing stress. That is the kind of stress that a rivet is subject to when the two plates it joins are pulling in opposite directions.

Strain is the result of a stress. If a stress of 30,000 lb. per sq.in. is applied to a bar or rod and the rod is lengthened a fraction of an inch, then the increase in length is the strain caused by the stress applied. These two terms are often confused. Stress is the cause, strain the effect.

Factor of Safety and Yield Point

Most operating engineers are familiar with ultimate or breaking stress, working stress and factor of safety. Briefly stated, ultimate stress is that stress which is sufficient to cause a rupture or break of the material under stress. The factor of safety is a factor determined by experience, which is divided into the ultimate stress to give the safe working stress. In the case of boilers the factor of safety varies from 4.8 to 5.5, which means that the safe working stress allowed is from $\frac{1}{4.8}$ to $\frac{1}{5.5}$ of the ultimate stress.

There is another property of the material that is important in its relation to safety of operation. That is the "yield" point. All materials when subjected to tensile stress, suffer deformation, or increase of length, the amount per unit of stress applied varying with each material. The ultimate tensile strength of boiler plate is approximately 60,000 lb. per sq.in. That is, if a tensile stress is applied to a piece of plate, when its magnitude reaches 60,000 lb. per sq.in. the material will break. But before that point is reached something else has occurred. The material of boiler plate, in common with other materials, possesses a certain amount of elasticity. Up to a certain point the material will, after the stress is removed, return to its original shape in exactly the same manner as a piece of elastic. When a certain stress is reached, this will no longer occur. The material will retain a permanent set, or increase in length in the case of tensile stress. Naturally, if the material has suffered a permanent increase in length, its other dimensions must have decreased. That is, there is a reduction in area to correspond with its increased length, the volume being the same. Any further stress applied will result in greater increase in length and greater reduction in area, until the metal breaks exactly as a piece of putty can be drawn out until it parts. The point where permanent deformation starts, or elasticity ceases, is called the "yield" point, and in the case of boiler plate is usually at half the ultimate stress, or 30,000 lb. per square inch.

A new boiler, built of first-class material and under first-class supervision to standard designs, is allowed a factor of safety of 4.8 by most authorities. This means that the material, having an ultimate stress of 60,000 lb. per sq.in. can safely bear a working stress of 12,500 lb. per sq.in. The boiler is completed and ready for the hydrostatic test, which is to be double the working pressure. That is, the stress during the hydrostatic test will be 25,000 lb. per sq.in. The yield point of the material is 30,000 lb. per sq.in. Therefore when the test pressure is on, the stress on the material is within 5,000 lb. per sq.in. of the point where permanent deformation will occur.

Where Danger Enters

It has been in the past, and is unfortunately at the present time, the practice of many engineers and some boiler inspectors to apply the hydrostatic test to determine working pressure. This is a distinct reversal of the proper method, which should be to determine from the proper working pressure, as found by calculation, the correct test pressure. As has frequently occurred in the writer's experience, the conditions are something like this: An owner has a boiler of uncertain age and condition, which has not been used for some time. He wishes to operate a sawmill or some other type of factory. He requires, we will say, 100 lb. pressure. An engineer or an inspector is called in, depending on the locality, and informed that 100 lb. pressure is required. "Very good," says this individual, "we'll just put 200 lb. water pressure on her. If she stands that, she is all right." Perhaps he is a bit conservative and thinks that 150 lb. should be enough. The pressure is applied, the boiler comes through with perhaps only a few slight leaks, which amount to nothing, and the owner is given a certificate for 100 lb. working pressure. Everyone is satisfied. Now what has happened? What is the actual condition of this boiler after the test is finished?

Let us come back to our piece of elastic. When it is new, it can be stretched and released, stretched and released, indefinitely, without any apparent effect. In course of time, however, it loses that property. If we tested a new piece of elastic till it broke, and then tested a similar piece that had lost its spring, we should find the old piece broke much easier than the new. Well, the material of the boiler is just like that elastic. Through years of use it has been expanding and contracting with every variation of steam pressure and condition of fire, so that it has lost some of the elasticity it had when new. Furthermore, its actual breaking limit has decreased to some extent, besides the yield point having been lowered. When it was new, the test pressure came within 5,000 lb. per sq.in. of the elastic limit. It is impossible to tell, without making careful measurements, whether the test made on the old boiler has not reached or exceeded the decreased elastic limit caused by years of work. If so, the boiler is in a condition where the shock caused by the sudden opening of a safety valve may be enough to cause a rupture of a plate. There is no question but that many boiler explosions could be traced to this cause.

A recent case in England, reported by the Board of Trade, where an explosion can be traced to this cause, will serve as an illustration. A vertical firebox boiler was examined by an insurance inspector, who reported thinning of the furnace plate in places. The working pressure was reduced. However, subsequent to this inspection some minor repairs were made and the boiler was tested hydraulically to 200 lb. by the boilermaker who made the repair. The safety valve was set for 75 lb., but a few days after the test, when steam was being raised, the boiler exploded, the pressure being only 35 lb. One man was killed and several injured.

We have shown that a hydraulic test may be in one case a pretty good guarantee and in another a dangerous experiment. The following points will serve as a guide to those who may be called upon to make tests on boilers that have been in service for any considerable length of time: Wherever it is possible to enter a boiler and examine it, this should be done first. The general condition should be noted and careful measurement of plates, pitch of rivets in joints, size of rivets, condition and diameter of stays, etc., all carefully noted. The plates and heads externally should be carefully examined. It is not the purpose here to explain the methods of inspection or what to look for, as that would take too much space, and it is presumed that anyone taking the responsibility of examining a boiler should be familiar with these points. The working presssure should then be calculated from the data obtained, and if there is any reasonable doubt of the condition, or if the boiler is over ten years old, the factor of safety should be increased to allow for this. If it is desirable after this examination to apply a hydrostatic test, the test pressure should not exceed one and one-half times the working pressure as determined by calculation.

The boiler should then be filled with water, taking great care to exclude all air, which can be done by opening a valve or cock on the highest point of the boiler or by easing back the safety valves to allow the air to escape as the water comes in. A steam pump should not be used for raising the pressure, as it is very difficult to control the increase of pressure. Serious shock may be sustained by reason of sudden raising of pressure by the stroke of a steam pump. A hand force pump should be used and the pressure gradually applied, careful examination being made after the working pressure has been reached. Leaks that develop should be carefully traced to their source, to make sure if they are from rivets or coming through the seams. After the test is made and the boiler emptied, a thorough examination should again be made internally and externally. In general, where a boiler has been under regular periodical inspection a test is necessary only after major repairs, to make sure of the workmanship of the job.

In the case of boilers that cannot be entered, such as small vertical firebox boilers and very small horizontal boilers, the hydraulic test forms the only means of discovering defects. In these cases the same general rules should be observed in applying the pressure, and as thorough an external examination as the circumstances may permit should be carried out during the test.

Conclusion

In general, hydraulic tests should never be carried out on boilers in the field except by men of experience and training in this class of work. By this, of course, is not meant the ordinary tests applied to water-tube boilers after an overhaul, the object of which is to ascertain if the caps and joints are tight.

Subjecting a boiler to an abnormal pressure, on the assumption that if it stands such pressure without showing weakness it can safely be worked at a lower pressure, is absolutely wrong. It would almost seem too obvious to write an article calling attention to it, but unfortunately, it is this method that is all too common. If this article causes some boiler owners and operating engineers to think a little more deeply on this subject, it will have performed a useful service.

Brown Electrical CO_2 Meter

A recent development of the Brown Instrument Co., of Philadelphia, Pa., is electrical CO_2 indicating and recording meters. The meter utilizes the method of gas analysis based on the difference in thermal conductivity of the various gases. A Wheatstone bridge is the means employed for comparing the thermal conductivity of the gas under test with that of another gas adopted as the standard of comparison.

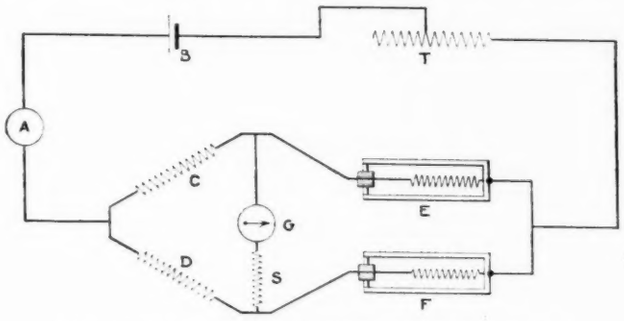


FIG. 1—DIAGRAM OF CONNECTIONS OF TESTER AND METER

Referring to Fig. 1, C and D are the ratio coils, while E and F are platinum spirals mounted in the gas chambers. Current from the storage battery B, controlled to a constant value by the rheostat T, subdivides and passes through the wire spirals. If the gases surrounding the two wires are alike in thermal conductivity, the cooling effects by conduction to the metal walls are equal. When this condition obtains, the spiral resistances attain the same temperatures and the bridge remains balanced. If, on the other hand, the gas under test differs from that used as a comparison standard, the bridge is thrown out of balance and the instrument G deflects accordingly. In practice one cell is filled with air and sealed for the comparison standard. The other cell is exposed to a flow of the flue gas under test.

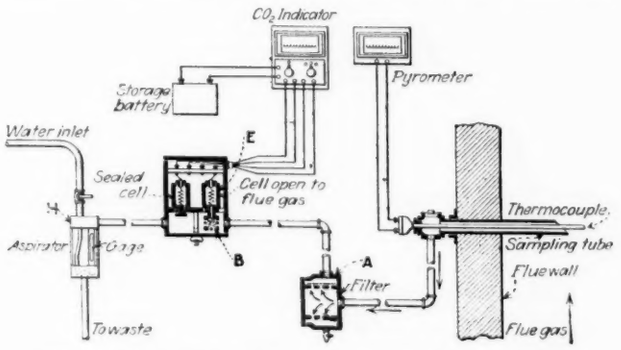


FIG. 2—GENERAL ARRANGEMENT OF CO_2 METER AND ACCESSORIES

The general arrangement of the meter and its accessories is shown in Fig. 2. The flue-gas sample is drawn in continuously by the aspirator H and passes from the sampling tube to the primary filter A, where any soot and moisture that have been carried through with the gas are removed. The gas then passes through the secondary filter B, Fig. 2, before entering the measuring cell E. The filter B contains a mixture of fine steel wool and glass wool, and its function is the absorp-

tion of any acid vapors that may be produced because of sulphur content of the fuel. From the cell *E* the gas passes to the aspirator *H*, which is equipped with a gage to indicate the draft in the gas line. The general arrangement of connections between the gas-cell assembly unit and the indicating meter is also shown in Fig. 2.

When it is desired to take the temperature measurement of the gas, the combined couple and sampling tube shown in Fig. 2 is used. This makes a compact installation and requires but one hole through the flue wall.

As the pipe used in connecting the meter with the sampling tube is comparatively small and the flow of gas through the cells is continuous, the time lag is practically eliminated.

Oil-Engine Spray Nozzles

BY H. F. SHEPHERD

One of the details of oil-engine spray valves that is of vital importance is the blowdown characteristic of the spray if the latter be of the spring-loaded differential-needle type. The term blowdown is used for want of a better term, as it is often applied to the operation of safety valves, which have properties in common with these sprays.

Referring to Fig. 1, which shows a spray of the Crossley type, let the large diameter of the needle be $\frac{7}{16}$ in., the diameter of the seat hole in the spray tip $\frac{1}{4}$ in. and the diameter of the seat itself $\frac{3}{32}$ in. Assume that the compression is 400 lb. per sq.in., the engine being a cold starter, and the spray pressure at the time of the lifting of the needle 4,000 lb. per sq.in.

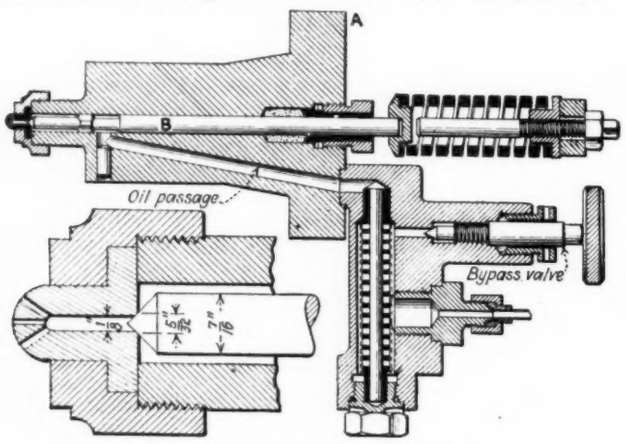


FIG. 1—TYPICAL OIL SPRAYER

The forces tending to lift the needle when it is on its seat are therefore 4,000 lb. per sq.in. acting on an annular area $\frac{7}{16}$ in. outside diameter and $\frac{3}{32}$ in. inside diameter. This area equals 0.13116 sq.in. and the lifting force due to the applied pressure equals 524.64 lb.

We are told by Rouleaux that a valve of this sort is acted upon by a pressure between the seating surfaces equal to the mean of the pressure on the inside and the outside of the seat. This may not apply strictly to very narrow seats. It certainly does apply to the broader seats, as was firmly impressed on the writer by an unfortunate experience with a highly ingenious differential pressure device which failed utterly in practice, owing to the working of this law. It is, therefore, included in the calculations, since it practically determines the blowdown of injectors with broad seats. Rouleaux, of course, assumes a liquid film between the seating surfaces which is acted upon transversely by the pressure outside the seat. Thus there will be another ring $\frac{3}{32}$-in. outside diameter by $\frac{1}{8}$-in. inside diameter, with a surface of 0.00690 sq.in. acted upon by the mean pressure between 4,000 lb. applied by the oil and 400 lb. compression pressure or 2,200 lb. per sq.in., resulting in a lifting force of 15.18 lb. And further, we have the compression pressure of 400 lb.

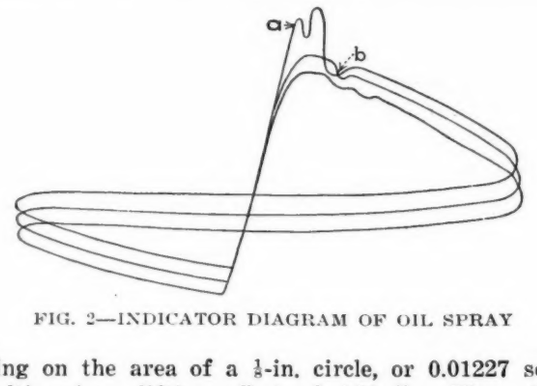


FIG. 2—INDICATOR DIAGRAM OF OIL SPRAY

acting on the area of a $\frac{1}{4}$-in. circle, or 0.01227 sq.in., resulting in a lifting effect of 4.91 lb. These three forces sum up to 544.73 lb., which is a suitable spring tension for this spray for lifting at 4,000 lb. per sq.in. oil pressure.

The instant the valve is off its seat, the oil pressure works against the full area of the $\frac{7}{16}$-in. circle of 0.15033 sq.in. and the needle is lifted to the stop. Before the needle can return to the seat, the pressure must fall to $\frac{544.73}{0.15033}$, or 3,624 lb. Thus theoretically, the spray seats when the pressure has been blown down to 376 lb. under the lift pressure. Greatly increasing the seat diameter will increase this drop of the pressure, very possibly causing the spray to deliver oil at velocities too low for suitable atomization.

By further and exactly similar calculation, using the pressure of 3,624 lb. per sq.in. acting on the needle after it is seated and 400 lb. per sq.in. prevailing at the spray tip, it may be shown that a force of 50.62 lb. unbalanced pressure holds the needle to its seat under this condition. This force of 50.62 lb. distributed over the seat area of 0.00690 sq.in. stresses the seat above the film pressure between its surfaces 7,336 lb. per sq.in. There is, therefore, very small chance of leakage if the seat is reasonably well made, regardless of whether the injector system is provided with a pressure-relieving bypass. The indicator diagram, Fig. 2, attached shows at *a* the point of spray opening. The rise above *a* is due to a rate of delivery greater than the orifices in the tip will pass at the lift pressure. Point *b* indicates closure of the spray needle. The slowly diminishing pressure after this point is due to indicator leakage and not to nozzle dribble. This is apparent from the action of the injector and it is also made apparent by the superimposed diagrams, which were taken by holding the injector needle to its seat with the stop preventing all discharge and adjusting the pump delivery so as just to make up for the indicator leakage. When new this indicator was tight but rather stiff. The diagrams are more satisfactory if the indicator leaks somewhat and is frictionless. The pressure scale of these diagrams is 3,000 lb. per in. The indicator is fitted with a piston of $\frac{1}{8}$ sq.in. area.

EDITORIALS

F. R. LOW, EDITOR

A. A. P. S. S. Conference on the Price of Coal

THE spectacular movements in the matter of our coal supply during recent years have fired the public, which gapes at the still-coming reports of the United States Coal Commission and, in an embarrassment of information, wonders what it is all about.

Such a conference as was staged at Philadelphia last week by the American Academy of Political and Social Science, offered a forum where the proponents of either side of the many questions involved presented their arguments for the consideration of the Ultimate Consumer. There were representatives of the operators and the miners and the public. The most dramatic session was that at which Governor Pinchot spoke from the same platform with S. D. Warriner, president of the Lehigh Coal and Navigation Company, with whom he had been exchanging courtesies in the public press. The Governor's address dealt with the questions:

Whether the increase which he granted the coal miners in the recent settlement was justified; and

Whether or not it ought to be passed on to the consumer.

All but two per cent of the workers were already on an eight-hour schedule. The experts of the Coal Commission advised him that the cost of extending it to all workers would be between one and two cents a ton. There was no question of the justice of the change. He recommended it and it was adopted.

The wage increase that he suggested was just and necessary to provide a decent American standard of living to match the going wages in similar industries and to compensate in part, but only in part, for the extra hazards of an industry in which twenty thousand men are hurt each year. The ten per cent increase was fully warranted, not only because a supply of anthracite, even at a small increase of cost, was infinitely better than no anthracite at all, but also upon its own merits.

The ten per cent increase would add about forty cents a ton on an average, but the steam sizes have to be favored because they sell in competition with bituminous, so that more than its own proportion of this increase is put on the domestic sizes. The largest increase in cost per ton chargeable to domestic sizes is fifty-five cents. In view of the increase due to the eight-hour day, Governor Pinchot had said to the public that this increased cost should not exceed sixty cents.

The least advance by the operators was seventy cents. Some added ninety cents, a few more than a dollar.

The much-talked-of Pennsylvania State anthracite tax requires a maximum addition of not over 12 cents a ton.

The report of the Federal Trade Commission says that on the week ended October 13 last wholesalers took margins reaching as high as $1.75 per gross ton (twelve times the average pre-war margins) and in the week ended October 20 as high as $1.50 per gross ton.

The gross margins of retail dealers in anthracite in Boston in November of last year ranged, according to the United States Coal Commission, from $4.07 to $6.22 a ton on stove sizes. In New England margins ranged from 13 cents to $6.05. In New York from $2.50 to $4.51.

It certainly looks as if there were several chances along the line for taking up sixty cents' worth of slack.

A Hydro-Electric Development Built on Sand

SCRIPTURE informs us that the man who built his house on the sand was foolish and suffered the penalty of having it blown down when the storm came. Be this fact or legend, modern engineering methods have provided means of locating great structures on sand and making them stay there. An interesting example of this is the fifty-thousand horsepower Sherman Island hydro-electric development of the International Paper Company, as described in the leading article in this issue. Most developments of this kind have solid rock foundations, but in this case the only solid rock is at the two extremities of the dam. This condition brought up many interesting problems such as supporting the dam on glacial deposits, preventing leakage under the dam, providing a spillway to take care of a flood of one hundred thousand cubic feet per second, making a canal laid in sand watertight and at the same time providing for expansion and contraction and other movements, location of the headworks, penstocks and power house on sand underlain with quicksand. These are the features that make this project stand out as an important development. Although this is not the only structure of the kind that has been built on sand, it is probably the largest development for the head involved, sixty-six feet.

The penstocks, which are of reinforced concrete, are designed to form a slab occupying the total area between the headworks and the power house, so as to reduce the bearing loads to a minimum. These have also been constructed to provide flexibility so that, although the power house, penstocks, headworks, canal and dam are one continuous concrete structure that is practically watertight, it is nevertheless flexible.

Another feature that is of particular interest to designing and construction engineers is the reinforcing of the draft-tube bells. Although some of the improved-type draft tubes possess desirable efficiency and operating characteristics, they present involved structural problems that increase their cost over some of the less-efficient types. New types of construction have been developed during the last year or two that are simplifying the reinforcing of the bell as well as the method of supporting it. Structural-steel reinforcing forms, as in this installation, seem to have much to recommend them. In other installations the draft-tube bells have been supported on columns similar to the stay vanes in the speed ring. A combination of these two designs would simplify not only the reinforcing problem but also that of supporting the turbine over the draft-tube chamber.

From the operating point of view one of the most interesting features of the plant is the provision made for obtaining a direct reading and a continuous record of the water going to the turbines. With the exception of some few installations on the Pacific Coast, little has been done toward obtaining a record of the water flow to hydraulic turbines. In some cases load curves are worked out for the turbines by tests and the flow to the turbines determined from these curves. Although this method has advantages that make it desirable in certain installations, in the average plant where meters can be used, they are preferable if the greatest output of kilowatt-hours is to be obtained from the water. It would in most cases be easier to provide means for checking the meter at stated intervals than it would be to test the turbines. The method of obtaining a record of water flow in the Sherman Island plant has certain untried elements in it, but expert engineering opinion is that there is no reason why it will not work successfully. In any event the results will be looked forward to with much interest.

Are Over-all Records Sufficient?

IN A RECENT editorial, a British contemporary emphasized the fact that "the main purpose of a boiler plant is simply and solely to produce the required amount of steam at the proper pressure and temperature with the least possible consumption of coal." From this the conclusion was drawn that the primary and the essential measurements in every boiler house are those of the coal burned and the water evaporated. It is hardly likely that anyone will dispute this statement, but it is easy to fall into the error of thinking that since these two measurements give the over-all result obtained, they render all other measurements of no value. Without saying so in such definite language, the editorial mentioned gives this impression. At first glance it would not seem unreasonable to ask, for example, why one should bother with a CO_2 recorder if the boiler is equipped with instruments for giving a continuous or daily record of the weight of steam produced and the weight of coal burned. What difference does it make what the CO_2 is as long as the plant gets results?

It takes only a little thought to make these questions answer themselves. Suppose a man is running a store. His main idea, presumably, is to make as much money as possible during the year. It might be assumed then that he would be quite satisfied with an accounting system that would give him each month two figures, the total amount of money he had put into the business and the total amount he had taken out, assuming for the sake of illustration that he is not enlarging or reducing his business either in equipment or inventory of merchandise. Yet a storekeeper who kept only such accounts would be the laughing stock of his fellow business men. Suppose the returns for a particular season happened to look pretty good. What possible way would there be for the storekeeper to tell from his accounts that they might not be still better? Suppose, on the other hand, his profits began to fall off or perhaps disappeared altogether. How could he tell where the trouble lay? He would know something was wrong, but could not remedy the situation until he knew what was actually taking place.

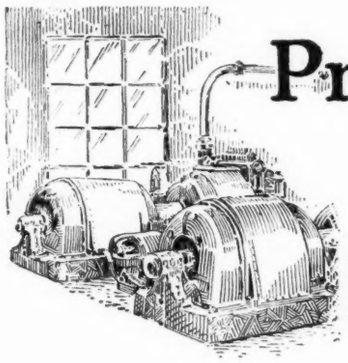
The analogy with the power plant is close. Suppose the coal and water records show that the plant is doing fairly well as plants go. How in the world is the operator to know that his particular plant might not do still better? If the efficiency falls off, how is he to improve it if he doesn't know where the loss occurs? It is impossible to overemphasize the importance of measuring the water and the coal, but this should not lead to underestimating the importance of the subsidiary records.

Ammonia Condensers

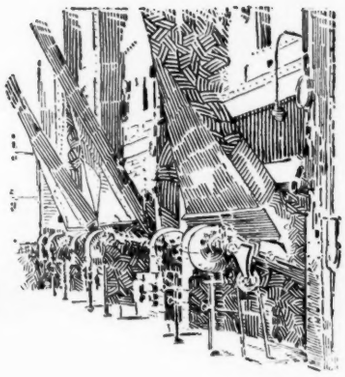
THE earliest condensers used in ammonia refrigerating plants were formed by making use of the most convenient material at hand; as a consequence, for many years the condenser was merely a series of pipes and return bends through which the ammonia circulated and over which the cooling water was splashed. The advantages possessed by this design were simplicity, ease of construction and practical freedom from leaks, since the joints, once made up, never needed to be broken. The low heat transfer of the atmospheric condenser led to numerous other designs, including the double-pipe, double-pipe counterflow, flooded and multiple-tube condensers, all these designs having a high coefficient of heat transfer but calling for the frequent opening of the joints for tube cleaning. But the smaller amount of cooling surface required, which decreases the floor space investment, has given them a decided popularity in the face of the objectionable joints.

In steam-plant practice condensers have never shown any similarity to those used in refrigerating plants. Since the condensing problem of both kinds of plants is essentially the same, being that of distributing cooling water in such a way as to take up the greatest amount of heat from the ammonia vapor or exhaust steam, it is surprising that the steam and ammonia condensers should not be almost identical, even though one must be built to withstand a pressure far above that of the exhaust steam. If the pipe coil is most economical for ammonia condensing, it should prove equally efficient when condensing steam even though the pressure conditions differ.

In the face of the similarity of problems the steam condenser is almost universally of the shell-and-tube design, and since steam condensers both in number and in capacity far exceed the ammonia condensers, it would seem that this design is superior to the pipe condenser. Refrigerating engineers realize this, and lately a number of refrigerating plants have installed shell-and-tube condensers and have found that there is an increase in heat transmission and a decrease in pumping head. Surprisingly, the refrigerating engineer has upturned the horizontal shell and tube, placing the tubes vertically. It is claimed that this gives an increased heat transmission by reason of the liquid covering the lower part of the tubes. On the other hand, in steam practice the presence of a film of liquid is assumed to lower the transmission coefficient. No doubt the actual advantage promised by the vertical tube is that of decreased pumping head, since the water falls through the tubes by gravity and the velocity is independent of the amount of water passing through the tube. While the shell-and-tube condenser has been applied only to large refrigerating plants, there is every reason to expect its more general adoption, to the advantage of the operator in eliminating the joint-leak trouble. That the pipe design still exists is probably due to the conservativeness of the refrigerating engineer.



Practical Ideas *from* Practical Men



Phasing Out the Generator for Parallel Connection

Some months ago the company with which I am connected decided to enlarge its manufacturing plant, and this in turn called for an extension to the generating plant, which consists at present of one 500-kw. unit. As the appropriation for this extension was somewhat limited, it was decided to buy a second-hand engine-driven unit of 300 kw. capacity from a neighboring city. This unit is due to be delivered in about six weeks, and in order to keep down expenses we intend doing the installation work without any outside help.

As I have never had any experience in synchronizing machines or operating them in parallel, I would be glad if some reader of *Power* would outline for me the method of phasing out the generator and also of connecting up the synchroscope or lamps. If lamps alone are all that is necessary, I would prefer to use them and save the cost of a synchroscope.

Our present unit is a three-phase 60-cycle 2,300-volt machine, and the only instruments on the switchboard are three ammeters (one on each phase) and a line voltmeter, and I understand the switchboard panel that is to come with the new unit is approximately the same.

We intend some time later to install an integrating wattmeter to register the combined output of the two units, so we should like some information as to what provision is necessary for this when connecting up the two machines to operate in parallel.

I shall appreciate some practical advice on the points I have outlined. E. C. CURTIS.

Cleveland, Ohio.

Trouble Experienced with a Duplex Pump

At a hospital where I was formerly employed as engineer, there was a compound duplex pump with a capacity of 500,000 gal. a day. The pump was normally operated at about half its rated capacity and discharged directly into the supply mains and also into a reservoir against a head pressure of 45 lb. The water supply was taken from Lake Ontario.

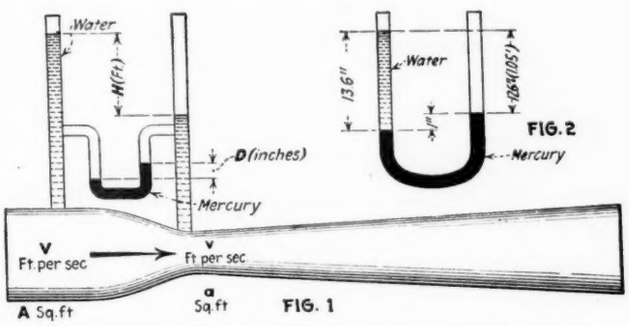
On one occasion we wanted to make some repairs to one of the pipe lines, so the pump was stopped and the valves on the discharge line leading to the reservoir were closed and those leading directly into the main were opened. When all was ready, the pump was started, but much to our surprise we were unable to build up any pressure on the main. Since the pump had been giving excellent service right along, this was somewhat puzzling. We made a hurried search for the trouble, as it was necessary to get the pump delivering water without delay, as the reservoir was of small capacity. The valves on the discharge were examined again and found to be in the right position. After trying the pump several times without success, we decided to change the valves back to their original position. Accordingly, the valves leading to the main were closed and those leading to the reservoir were opened. When steam was again given to the pump, it worked as well as ever.

We concluded the trouble to be due to the fact that the discharge valve springs were old and not strong enough to seat the valves when working against practically no pressure, but with a head pressure from the tank the valves seated all right. That this was the cause of the trouble was proved later, as the pump worked perfectly with an open discharge when new springs and valves were installed. J. E. NOBLE.

Toronto, Ont., Canada.

Velocity of Discharge Through Venturi Tube

In using a formula for velocity of water through a venturi tube, it should be borne in mind that in the usual formula, $v = \sqrt{2gH}$, for the venturi (where v = throat velocity, feet per sec., and g = acceleration of gravity), the value of H is the difference of head of



MEASUREMENT OF DIFFERENTIAL PRESSURE IN VENTURI TUBE

liquid discharged through the venturi tube as would be shown by comparison of two piezometer tubes, one connected with the inlet and the other with the throat of the venturi tube, as illustrated in Fig. 1.

If we take a U-tube containing mercury, and into the left branch pour water (at 62 deg. F.) until it stands 13.6 in. above the mercury, then, as shown, Fig. 2, the mercury will rise in the right branch until there is exactly 1 in. difference in mercury levels between the two branches; but, as shown, the difference in head

of water in the two branches is only 12.6 in. In other words for each inch difference in mercury levels in the U-tube the corresponding differential head of water is 1.05 ft. Hence, where the mercury and D is the difference of mercury level, the formula

$$v = C \sqrt{\frac{1}{1 - \frac{a^2}{A^2}}} \times \sqrt{2gH}$$

for velocity at the throat section, where v = velocity at the throat section in feet per second, a = area of throat in square feet, A = the area of the entrance section, and H = the difference of head in feet, does not, as given in a previous issue of *Power*, become,

$$v = \sqrt{\frac{1}{1 - \frac{a^2}{A^2}}} \times \sqrt{2gD},$$

where D is taken as inches of mercury column, equal to H feet head of water, since $H = 1.05D$, and the formula becomes

$$v = C \sqrt{\frac{1}{1 - \frac{a^2}{A^2}}} \times \sqrt{2g(D \times 1.05)}$$

$$= 8.22C \sqrt{\frac{1}{1 - \frac{a^2}{A^2}}} \times \sqrt{D}$$

or better,

$$v = 8.22C \sqrt{\frac{r^2}{r^2 - 1}} \times \sqrt{D}, \text{ where } r = \frac{\text{inlet area}}{\text{throat area}} = \frac{A}{a}$$

Providence, R. I. CHARLES G. RICHARDSON.

The Use and Abuse of Belts

When I was a young engineer, I had charge of the power and machinery in a building that had a custom sawmill on the ground floor, a planing mill on the second and brush machines on the third. Power was transmitted on the main drives by rubber belts, while leather belts supplied the smaller transmissions. All the belts were durable and rendered satisfactory service, although the belt dressing that was used exclusively is not recommended as a panacea or universal remedy. We bought it at the local gas works; it was called tar, a byproduct of the manufacture of illuminating gas. It had two good qualities—it was cheap, and it made the belts drive almost as positively as gears for a short time. The negative quality consisted of great lumps of tar on the faces of the pulleys, that had to be removed from time to time. Also an intolerable noise was always in evidence directly after the application of tar.

The main belt was 18 in. wide and six-ply in thickness, supported by an idler near the driven pulley. As it was not possible to make such a belt endless, it was laced with heavy leather lacing. For some unaccountable reason it had to be laced in two places (about 15 in. apart) and when the lacings passed one pulley and up over the other the vibration was severe. After I had been in charge of the plant for some time, I removed the idler, and the main belt carried the full load without slipping, and the noise was greatly reduced.

The 54-in. saw was driven by a 14-in. rubber belt that passed over two idler pulleys. It was necessary for the men, as they moved about the mill, to pass under this belt, hence it had been raised out of its natural path. This double idler was adjustable to keep the belt at the proper tension.

All belts which were 8 in. or less in width were of leather. In one case the driving pulley was directly over the driven and the belt was very loose when the saw that it drove was not in service. The belt could be stopped by running a board on the saw, which could then be removed for filing. This belt was running practically all the time the engine was in operation, hence if the pulley was stopped more than a few minutes at a time, it would burn the belt. When the saw was to be used, a hinged idler was let down on the belt, which made it tight enough for any service. Metal hooks were used for joining the ends together.

When the business became more prosperous, we bought some belt dressing that was supposed to be a big improvement over the tar. It was a thick yellow paste, that caused the belts to stick to the pulleys, but it was less tenacious than the tar. It certainly was more expensive, and in my opinion for that particular place the tar answered every purpose.

When I left that place I took charge of an engine where a leather belt was used to transmit the power. I worked there for nearly one year, and during that time nothing was put on the belt. The next plant to be mentioned included a 24-in. leather belt to transmit power from the flywheel to the jackshaft. The belt was two-ply and never showed a break after many years of hard service.

One day a neighboring mill was destroyed by fire, and we were called upon to do the work of both mills. I started the engine an hour earlier in the morning, made no stop at noon, and kept the machinery in motion four hours longer than usual at night. We used plenty of belt dressing, so as to prevent all loss by slipping of the belt.

This belt worked to one disadvantage, as the load was carried by the upper part of it. The engine ran "over" and the jackshaft was nearly above the cylinder head. However, there was one advantage, as the jackshaft was about 15 ft. above the cylinder, thus causing the belt to stand at an angle of about 45 deg. A good share of the weight of the belt was carried by the smaller pulley.

On one occasion we had a fire in the mill. It started in one of the bearings of the jackshaft. All the pulleys and belts were covered with dust from the planers, hence when the fire started, it followed this light dust rapidly. The engine was stopped as soon as possible, and when the fire department finished their operations, the belt was examined and found to be in perfect condition. Even the ¾-in. leather lacings were not damaged.

My next experience was in a silk mill, in which two lines of shafting extended through the entire length of the main building. These were about 8 ft. apart, with a pulley on the end of each in the engine room. To meet this condition the flywheel was turned for two belts. The engine was set to run "under" and the lower part of each belt carried the load. They gave excellent service, and no belt dressing was ever used on them.

From this silk mill I went to another saw and planing mill where the engine was overloaded when all the machines were in operation. Here there was no jackshaft, as the mainshaft extended into the engine room and it carried the main pulley. The crankshaft was higher than the mainshaft, and as the latter ended near

the cylinder head, the engine running "over," the load was carried by the upper part of the belt. Sometimes this belt would be tight enough when the mill was shut down on Saturday night, but would be quite loose on Monday morning. As the lower part was slack, it would touch rough mason work, which could not be easily removed, and the belt would be scratched until tightened sufficiently to clear. The mill always ran until noon, and then the belt was shortened. The temperature of this room frequently was 110 deg., but this did not affect the belt. Conditions made it difficult to apply belt dressing, and this was one reason why not much of it was used.

The next plant was a large machine shop that had two 10-in. main belts, one in either direction from the flywheel, which was 12 ft. in diameter. Each of the main pulleys was fitted with a friction clutch, therefore it was possible to run one part of the shop if the whole was not wanted. It was customary to pull these clutches out before starting the engine in the morning. When the cylinder was well heated, the clutches were slowly pulled in, thus starting all the machinery. Belt dressing was used freely on these belts, hence when it was necessary to shorten them to prevent slipping, it was always a disagreeable job.

One day I visited a local paper mill. I saw an 18-in. rubber belt that was very tight, and it was also slipping badly. I asked the superintendent if he wanted to have the slipping stopped, to which he replied in the affirmative. I took some liquid belt dressing, heated it and applied a small quantity. In a few minutes all slipping had stopped, and the machine was driven at full speed.

I met one of the engineers soon after, and asked him how he liked the belt dressing that the superintendent purchased. His reply was that it was of no value. Further investigation showed that they had poured about a quart of the belt dressing on the face of the flywheel, causing the belt to come off the flywheel, and it was hard work to get it on again.

New Haven, Conn. W. H. WAKEMAN.

Increasing the Speed of Air Compressor Reduces Capacity

An experience I had some time ago, when taking charge of a plant in which there were two compressors of the same size and make, may be of interest to other readers. The compressors were compound with 10- and 16- by 16-in. steam and air cylinders and were rated at 550 cu.ft. of free air per minute running 135 r.p.m. One machine had a 10-in. air intake on the low-pressure side, while the other had only 6-in. air intake. The machine with the 10-in. intake would deliver its rating when operating at 125 r.p.m., while the other one would not deliver anywhere near its rated capacity.

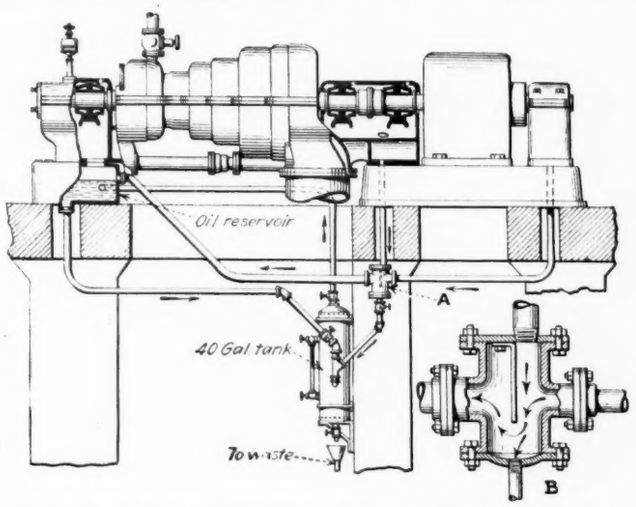
The master mechanic thought he would increase the output of the latter machine, so he had the governor pulley lagged sufficiently to increase the speed of the compressor from 135 to 165 r.p.m. This resulted in the machine's doing even worse than before. It was then decided to try the compressor at a slower speed, and this gave much better results. With the compressor operating at 165 r.p.m. and with the usual amount of air being drawn from the receiver, 60 lb. pressure was all that was possible to get. By reducing the speed to 100 r.p.m. the pressure increased to 80 lb., and by reducing it still further to 90 r.p.m. we got 100 lb. pressure. From these results it was evident that the 6-in. air intake which had an area of approximately one-seventh that of the low-pressure piston, was much too small to supply the cylinder at the high speeds, but by reducing the speed the interval between strokes was lengthened, allowing more time for air to pass into the cylinder. J. M. FRANKLIN.

Buffalo, N. Y.

Method of Removing Water from the Oil on Turbine Units

Some months ago we began to have trouble from the water not separating properly from the oil on our 2,000-kw. turbine, and it may be of interest to other readers to know how we overcame the trouble.

From tests made on the oil by dipping a bottleful from the oil reservoir and allowing it to stand, it was seen that the water could be effectively removed by providing a storage tank or reservoir where the oil



HOW SEPARATING TANK WAS CONNECTED TO OILING SYSTEM OF TURBINE

would be practically at rest and thus allow the water to separate and be drained away as required.

Accordingly, a 40-gal. domestic hot-water tank was obtained and fastened to the turbine foundation. The general arrangement of the return oil piping is shown in the illustration. The only change made in the oiling system was to replace the tee in the return-line connection to the center bearing with a reducing cross, as at *A*. The connection from the bearing was offset and a baffle attached to the flanges as shown by the dotted lines and also in detail at *B*. The bottom of the tee and also the drain from the oil reservoir on the turbine were connected to the tank as shown.

A connection from the top of the tank was led back to the oil reservoir at a point slightly below the normal oil level. A gage glass was fitted to the tank so that the amount of water could be observed and the draining done at the proper intervals. The duty of testing the oil and draining the tank is left to the operator on the day shift.

Since this change was made, the water has separated readily from the oil and it is only necessary to change the oil in the turbine every six to eight months, whereas it was formerly changed about every three weeks.

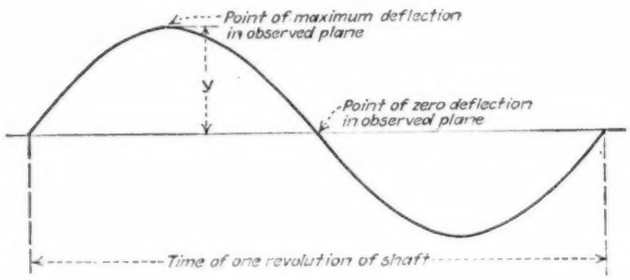
Memphis, Tenn. W. H. GREEN.

Comments from Readers

Location of Weights in Turbine Balancing

In the Oct. 16 issue L. Long presented an interesting article on "Correcting Vibration in Reaction Type Turbines." In the first paragraph, on page 612, he explains that the throw of a weight is not directly away from the center of an axis because a weight turned loose from the periphery of a wheel flies off at a tangent. This statement is made to explain why, in balancing large rotating bodies, the weights should not be added to a point opposite the point of heaviest marking.

In my opinion Mr. Long's explanation is incorrect; even if a weight turned loose from a wheel flies off at a tangent direction, the centrifugal force due to an unbalanced mass in a rotor is always in a radial direction. When a spindle is operating at a speed below the critical, as is the case in most reaction turbines, the point of maximum deflection on the circumference of the shaft


Point of maximum deflection in observed plane
y
Point of zero deflection in observed plane
Time of one revolution of shaft

DEFLECTION OF SHAFT PLOTTED AGAINST TIME

will always be coincident with the center of gravity of the unbalanced mass. Above the critical speed the point of maximum deflection will be 180 deg. away from the center of gravity of the unbalanced mass. The reason the weights should not be placed directly opposite the point of heaviest markings when operating under the critical speed is because the point of heaviest markings does not always correspond to the point of maximum deflection. If the deflection of a shaft when vibrating is plotted against time, it results in the sine curve shown in the illustration. From a study of this curve it will be seen that when a pencil is held to a rotating shaft, the point of heaviest marking will always occur before the point of maximum deflection has come into contact with the pencil.

With spindles turning at high speeds the point of heaviest marking may be even more than thirty degrees in advance of the point of maximum deflection, depending upon how the markings are obtained. If the pencil is held toward the shaft with much force, it will come into contact with the shaft at a point considerably in advance of the point of maximum deflection. Upon making contact and acquiring the velocity due to the change in deflection of the shaft, the weight of the pencil and that part of the hand that moves with the pencil may be so great that the pencil will break contact with the shaft even before the point of maximum deflection is reached. In this case the point of heaviest markings is largely determined by the manner in which the pencil is held.

J. E. RICHARDSON.

Pittsburgh, Pa.

Pump Pounds from Too Much Suction Lift

In the Oct. 30 issue R. A. Cultra gives reasons for a pump pounding. Some of them are new to me, and one in particular—a leaking check valve in the feed line—causes the pressure to fall on the discharge valves and will result in pounding. According to my understanding of pump operation this pressure is unavoidable. A pressure gage on the discharge line will show that the pressure on the discharge valves is practically constant. During the suction stroke the unbalanced pressure is the feed-line pressure plus that due to the suction lift, plus the spring tension, while during the discharge stroke the pressure is practically balanced, the pressure below the valve being slightly greater than that above.

Mr. Cultra has omitted one certain cause for pounding, particularly with pumps of the outside-packed plunger type; that is, worn plungers. The plunger is driven into the stuffing box until the shoulder jams in the packing. When steam is admitted to the other side of the piston, it requires a considerably higher pressure to start the plunger than is necessary to keep it moving after the shoulder has moved out of the packing. The result is that the plunger starts with a jerk and breaks the water column. This causes a pound and may even drive the steam piston against the cylinder head.

As there is more inertia to overcome in the case of a high suction lift or a long horizontal line than when the suction is short, it will be easily seen why pounding is more likely to occur in some cases than in others. When the plungers are in good condition, the remedy for pounding is to install an air chamber in the suction line near the pump, also to have the steam valves so set that before a plunger comes to rest, the opposite one is in motion. When there are shoulders on the plunger, the obvious thing to do is to remove them.

Recently, we overhauled a duplex boiler-feed pump on which the valves were leaking badly. The seats had become crowned owing to wear. As the temperature of the feed water was from 210 to 220 deg., a hard composition valve was necessary. If at any time the temperature dropped considerably, the valves would not conform to the seats. We trued up the seats and valves, also the plungers, which were somewhat worn and had shown a slight tendency to stick in the stuffing boxes at the beginning of the stroke.

Several engineers have expressed the opinion that we would have trouble pumping such hot water with a

duplex pump, but so far we have not experienced any, possibly due to the fact that we keep the tension on the springs as light as possible, practically negligible. A common remedy for pump trouble is to make the springs stiff. I believe this is a mistake, especially with the springs on the suction valves.

Toronto, Ont., Canada. R. McLAREN.

Does a Partly Closed Feed Valve Increase the Duty on the Feed Pump?

The reply of Mr. McLaren in the Oct. 23 issue to the inquiry of Mr. Harris in the Sept. 18 issue "Does a Partly Closed Feed Valve Increase the Duty on the Feed Pump?" may be misleading to many readers.

Mr. McLaren's statement, that a partly closed feed valve does increase the duty on a feed pump "and the amount that the duty is increased depends on the increase in pressure," would be true providing that the pump would deliver the same amount of water at different heads, but he ignores the fact that the capacity of a centrifugal pump varies with the head according to the characteristic curve.

Assume a boiler pressure of 95 lb. and a loss of 5 lb. in the feed line due to friction, when the feed valves are open. If the feed valves are closed sufficiently to raise the pressure to 25 lb. above the pressure necessary to feed the boilers with the valves wide open, and the pump is delivering 1,000 lb. of water per minute, as stated in the article referred to, the work done by the pump would equal 1,000 × 125 × 2.3 = 287,500 ft.-lb. (2.3 being the height in feet of a column of water equal to 1 lb. pressure per square inch).

Now refer to Fig. 1 and assume that the curve shown is the characteristic curve of the pump. When the pump is delivering 1,000 lb. of water per minute, *AB* will represent the head on the pump due to boiler pressure, *BC*, the head due to pipe friction and *CD* the head due

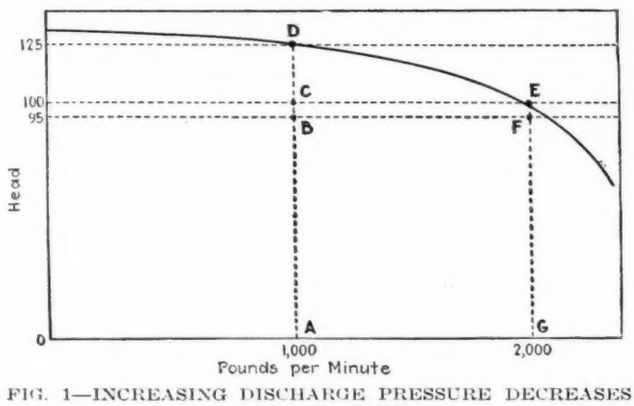


FIG. 1—INCREASING DISCHARGE PRESSURE DECREASES DUTY ON PUMP

to partly closed feed valves. Now suppose that the feed valves are wide open; then *EF* will represent the head due to friction in the line and *FG* the head due to boiler pressure, and the pump will deliver 2,000 lb. of water per minute and the work done will equal 2,000 × 100 × 2.3 = 460,000 ft.-lb. Therefore, by increasing the head on the pump 25 lb. by partly closing the feed valves, the duty on the pump would be decreased 460,000 — 287,500 = 172,000 foot-pounds.

Fig. 2 shows a different type of characteristic curve for a boiler-feed pump taken from "Gelhardt's Steam Power Plant Engineering." In this pump an increase in head would mean an increase in capacity up to a certain point.

The principles involved are explained very clearly by R. K. Annis, in an article entitled, "Characteristic Curves of Centrifugal Pumps," in the Oct. 23 issue of *Power*.

GEO. E. GASTER.

Tyrone, Pa.

Replying to the article by A. L. Harris in the Sept. 18 issue, the steam pressure in the boiler determines the pressure that must be built up in the pump before any water will be delivered to the boiler, and any restrictions placed in the feed line between the pump and the

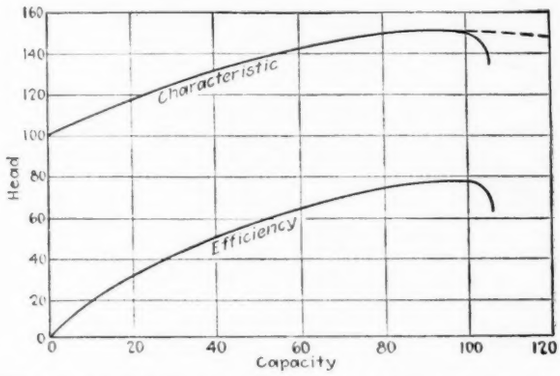


FIG. 2—CENTRIFUGAL PUMP CHARACTERISTIC FOR BOILER FEEDING

boiler, such as a partly closed feed valve, would add to the work of the pump to deliver the required amount of water to the boiler. A partly closed feed valve would necessitate a higher pressure in the feed line in order to drive the water through the restricted opening at a higher velocity, hence it would require more work to be done by the pump.

For example, a pump with a 4-in. diameter water cylinder and a 6-in. diameter steam cylinder would have a ratio of areas of 12.56 sq.in. on the water end to 28.27 sq.in. on the steam end. The total pressure on the water piston of the pump would be 12.56 multiplied by the pressure of 150 lb., or 1,884 lb., and this divided by the area of the steam piston would give a mean effective pressure of 66.6 lb. As a duplex pump does not get the benefit of the expansion of the steam but exhausts the steam at this pressure, any excess pressure set up in the water end would naturally call for a higher pressure in the steam end and mean just that much waste of steam. I have not gone into the efficiencies of the pump, nor have I considered the pressure necessary to overcome the friction and to force the check valve open against the steam pressure, but in my experience I have found this to be from 10 to 25 pounds.

The greatest cause for vibration is usually found in the condition of the pump; namely, leaky valves, water plungers leaking, or one plunger coming to rest before the other one starts on its stroke, which allows the check valves to seat before the other plunger builds up a pressure.

F. L. BAILEY.

Coffeyville, Kan.

One great use for the Orsat is in locating air leaks. If the carbon dioxide in the uptake is much lower than that in the first pass, it proves that substantial air leaks exist between these points.

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Motor for Geared Well Pump

What size of electric motor would be required to operate a geared deepwell pump discharging 40 gal. of water a minute against a total head of 190 ft., through 328 ft. of 2½-in. pipe with one check valve, one gate valve and six long-sweep elbows? L. J. G.

Allowing the loss of head due to fittings is the same as for 25 ft. of 2½-in. pipe, the loss of heat from pipe friction would be the same as for 328 + 25 = 353 ft. of 2½-in. pipe. For discharging at the rate of 40 gal. per min., the loss of head in 2½-in. pipe would be about 22 ft. per 1,000 ft., or 22 × 0.353 = 7.76 ft. head for 353 ft., making the head pumped against equivalent to 190 + 7.7 = 197.76 ft. head; and the water horsepower would be 40 × 8.33 × 197.76 ÷ 33,000 = 1.99, or practically 2 hp. Taking the mechanical efficiency of the pump at 60 per cent, the actual power required of a motor would be 2 ÷ 0.60 = 3.33 hp., and assuming operation of the motor with 70 per cent efficiency the input required would be 3.33 ÷ 0.70 = 4.75 electrical horsepower.

Power Required To Compress Natural Gas

What number of horsepower would be required per revolution per minute to compress natural gas from 50 lb. initial gage pressure to 290 lb. discharge gage pressure in a compressor cylinder 15¾ in. in diameter and stroke 42 in. with piston rod and tail-rod 3½ in. in diameter? E. O. M.

For natural gas the value of the exponent n in the compression curve PV^n = a constant, is substantially the same as for ammonia, namely, $n = 1.3$.

Formulas for finding the work of ammonia compression with allowance for cylinder clearance are given pages 418-20, Sept. 10, 1922, issue of *Power;* also a table of m.e.p. constants per pound of suction pressure for different ratios of compression and for various percentages of cylinder clearance. For an initial pressure of 50 lb. gage, or 65 lb. absolute, and compression carried to 290 lb. gage, or 305 lb. absolute, the pressure ratio would be $\frac{P_B}{P_C} = \frac{305}{65}$ = about 4.7, and by interpolation from the table, the m.e.p. per pound of initial pressure would be 1.73 lb. for 3 per cent clearance, 1.69 lb. for 4 per cent clearance and 1.6 lb. for 5 per cent clearance. Hence for 65 lb. absolute suction pressure the m.e.p. would be 65 × 1.73 = 112.45 lb. m.e.p. for 3 per cent clearance, 65 × 1.69 = 109.85 lb. m.e.p. for 4 per cent clearance and 65 × 1.6 = 104 lb. m.e.p. for 5 per cent clearance.

With a cylinder 15¾ in. in diameter and piston rod and tailrod 3½ in. in diameter, the net area of the piston would be [(15¾ × 15¾) − (3½ × 3½)] × 0.7854 = 185.21 sq.in.; and with 42 in. stroke and one revolution per minute the power per pound m.e.p. would be $\frac{42 \times 185.21 \times 2}{12 \times 33{,}000}$ = 0.03928 hp. Hence with 3 per cent cylinder clearance the power per revolution per minute required for adiabatic compression would be 0.03928 × 112.45 = 4.417 hp.; with 4 per cent clearance, 0.03928 × 109.85 = 4.315 hp. and with 5 per cent clearance, 0.03928 × 104 = 4.085 hp. The gross power, including friction of the compressor, would be about 15 per cent more, depending on the efficiency of the compressor.

Location of Weights for Balancing Steam Turbines

Should weights for balancing steam turbines be placed opposite the high marking? T. E. H.

In balancing steam turbines it is common practice to begin by marking the shaft with a pencil, after having painted it with chalk and alcohol or similar compound. In some cases the high marking on the shaft indicates the heavy side of the rotor and weights should be placed opposite the heavy markings. In other conditions of turbine operation the heavy markings actually indicate the light side of the rotor, so that weights should then be placed somewhere near the same side of the shaft on which the heavy markings occur. The fact that both conditions occur in different machines can perhaps be understood from the fact that we have to deal with two conditions in a rotating disk or other types of rotors. In the first place there is centrifugal force which naturally would pull the heavy side away from the center; in the second place there is a tendency for the disk, under certain conditions, to seek its own center of gravity and rotate about this axis of gravity.

The first condition is easily understood. In regard to the second, if we take a pasteboard disk and cut it round and place a weight near the periphery, the disk will tend to rotate around the heavy weight rather than around its own center of symmetry. If we free the disk with a spinning motion, we will see that it tends to rotate around the weight itself or some point very near the weight. Under such conditions the high markings would be on the side opposite the weight. In order to balance such a disk, weights would have to be placed somewhere on the same side of the center of the disk as the high markings occur. This would be exactly opposite from the first condition where centrifugal force only is obvious.

In steam turbines it seldom happens that the weight should be placed exactly opposite to heavy markings. With a very flexible shaft, where the turbine operates above critical speed, the tendency is for the rotor to

revolve about its center of gravity rather than its center of symmetry. This corresponds to the loaded pasteboard disk, and weights should be placed somewhere near the same side that markings occur.

On the other hand, where a rigid type of shaft is used, which operates considerably below the critical speed, we have the first condition—namely, that of centrifugal force—as the stronger condition to deal with. Here the weights must be placed practically opposite the heavy markings.

In ordinary practice weights seldom go exactly opposite or exactly with markings—they usually go at some angle, either to one side or the other, depending on the type of machine and how far above or below critical speed it is operated.

It seldom happens that accurate balancing can be done by marking the shaft alone. Accurate final balancing is almost always done by feeling the vibration and judging the effect of various combinations of the weights. Balancing machines for locating the positions and amounts of weights required for balancing the rotor have been described in previous issues.

Inertia of Small Piston with High R.P.M. of Engine

What would be the inertia of a piston weighing 3.5 lb. where the stroke of the engine is 4 in. and speed 2,000 r.p.m.; also at 1,800 r.p.m.? M. E. W.

The inertia of a piston and connected reciprocating weights, at the end of the stroke, is equal to the centrifugal force that these weights would develop if they were concentrated and rotated around a center with radius equal to the crank radius of the engine and revolved at the same r.p.m. as the crank of the engine.

Centrifugal forces $= 0.00034084\ WRN^2$

where W = weight in pounds, R = radius in feet and N = number of revolutions per minute.

With 4-in. stroke, $R = 2$ in. $= \frac{1}{6}$ ft., W, the weight of piston, $= 3.5$ lb., and $N^2 = 2{,}000 \times 2{,}000 = 4{,}000{,}000$, then the inertia of the piston alone at the end of the stroke would be $0.00034084 \times 3.5 \times \frac{1}{6}$ ft. $\times 4{,}000{,}000 = 795.29$ lb. Where $N = 1{,}800$, $N^2 = 3{,}240{,}000$, and the inertia would be $0.00034084 \times 3.5 \times \frac{1}{6}$ ft. $\times 3{,}240{,}000 = 644.18$ pounds.

Cause of Direct-Current Generator Not Building Up Its Voltage

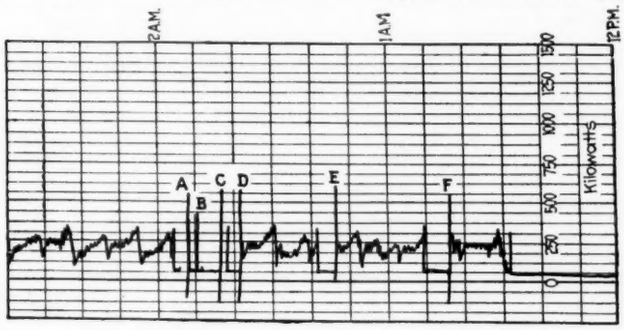
A 50-kw. 230-volt compound-wound direct-current generator that was operating satisfactorily in one location, was moved to another plant and now will generate only about 5 volts. When a water rheostat is used to load the machine, the current will increase to about 70 amperes before the electrodes in the rheostat come in contact with each other, when the circuit breaker will trip. Why is it that the generator will not build up its voltage without a load, and what causes the circuit breaker to open? A. A. W.

Since the machine will generate on the series winding, this winding is connected in the right relation to the armature, otherwise the generator would not build up voltage as a series machine. Therefore this ties the cause of the trouble down to the shunt-field circuit, which may be opened by a break in the wiring or in the field windings or rheostat. The field windings may be short-circuited out of circuit by connecting both field terminals to the same side of the armature. Or the field windings may be connected in the wrong relation to the armature, in which case the field leads should be interchanged to their terminals. If the machine's rotation is the same in the new location as in the old, and the connections are correct, then there is no reason for its not building up. If the direction of rotation was changed, shifting the brushes one pole space around the commutator will give the correct relation of connection for the machine to build up its voltage. This can also be accomplished by interchanging the armature leads or interchanging the field loads.

Calculating Power Taken by Motors from Recording Meter Chart

In the figure is shown part of a curve taken from one of our recording wattmeters. This meter is recording the power taken by a 250-hp. motor driving a variable load of rubber mills. In addition to this motor there is a 57-hp. motor and a number of others ranging in size from 2 to 10 hp. The power is 440-volt 60-cycle three-phase. The peaks *A*, *B*, *C*, *D*, *E* and *F* show when the oil switch was tripped. How is the horsepower and current load that was on the system when the oil switch opened figured from this curve? The power factor is approximately 0.80. J. G. W.

Take for example peak *E*, which has a value of about 600 kw.; the horsepower load is equal to kilowatts $\times$



LOAD CURVE FROM RECORDING WATTMETER

$1{,}000 \div 746 = 600 \times 1{,}000 \div 746 = 804$. This is the electrical horsepower supplied to the motors when the switch opened. The mechanical horsepower work done by the motors will depend upon their efficiency and will vary according to the load on the motors, their sizes and method of drive. If an efficiency of 85 per cent is assumed, the mechanical horsepower load driven by the motors is $804 \times 0.85 = 683$.

The current in a three-phase circuit, where the voltage, power factor and kilowatts are known may be found by the formula,

$$I = \frac{Kw \times 1{,}000}{E \times P.F. \times 1.732} = \frac{600 \times 1{,}000}{440 \times 0.80 \times 1.732} = 985\ amp.$$

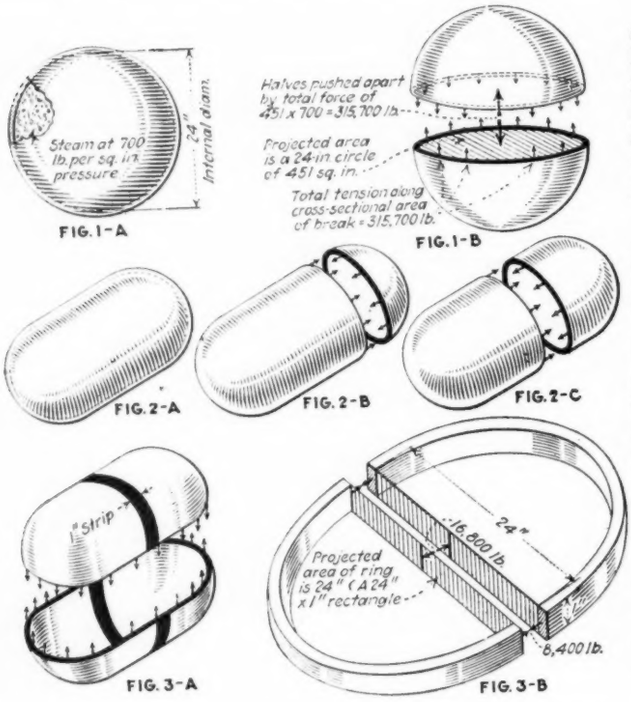
If the switch was tripped by a sudden overload on the system, it is probable that the current peak was much higher than indicated by the motor, since it is not sensitive enough to follow a sudden change in load. Another thing, the power factor is undoubtedly different under overload than for normal conditions. However, the peaks give a fairly accurate idea of what happened in the circuit and show that the switch opened owing to an overload and was not tripped accidentally

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications.—Editor.]

Easy Lessons in Engineering

Stresses in a Shell Under Pressure

The word "shell" in the title is used to mean any pressure vessel, such as a steam pipe, a compressed-air receiver, the shell of a horizontal return-tubular boiler or the drum of a water-tube boiler. For the sake of simplicity, it will be assumed at first that the tanks are seamless. Then account will be taken of the riveted joints that are necessary in most cases. The simplest tank to figure would be a seamless spherical shell as shown in Fig. 1. If the material were perfectly uniform and the pressure were increased until the tank burst, the fracture would be a ring running around the sphere at its greatest diameter, but in some unknown direction. The fluid pressure of 700 lb. per sq.in. acts equally in all directions, but assuming a fracture along a particular plane, the force to be considered is the total perpendicular to that plane. The case is exactly like that of a piston with a curved end. No matter what the shape of the end of the piston, the projected area is always the same as the area of the bore of the cylinder. The area to be considered here is that of a 24-in. circle, or 451 sq.in., and the total force tending to push one half away from the other is 451 × 700 = 315,700 pounds.


Steam at 700 lb. per sq. in. pressure
24" Internal diam.
FIG. 1-A
Halves pushed apart by total force of 451 x 700 = 315,700 lb.
Projected area is a 24-in. circle of 451 sq. in.
Total tension along cross-sectional area of break = 315,700 lb.
FIG. 1-B
FIG. 2-A
FIG. 2-B
FIG. 2-C
1" strip
Projected area of ring is 24" (Δ 24" x 1" rectangle
24"
16,800 lb
6,400 lb.
FIG. 3-A
FIG. 3-B

FIGS. 1 TO 3—RESISTANCE OF SHELLS TO LONGITUDINAL AND CIRCULAR RUPTURE

This is resisted by all the metal in the ring (see *B*, Fig. 1). The area of this ring is its width, $\frac{5}{8}$ in., multiplied by its average circumference. Since the shell is thin in proportion to its diameter, the inner circumference may be used without serious error. This is 24 × 3.1416 = 75.5 in., so the metal area is 75.5 × $\frac{5}{8}$ = 47.1 sq.in. Therefore the unit stress, or tension on each square inch, is 315,700 ÷ 47.1 = **6,700** lb. per square inch.

Next take the case, illustrated in Figs. 2 and 3, of a seamless cylindrical shell with hemispherical ends. Here the diameter is 24 in. inside and the shell thickness $\frac{5}{8}$ in. as before. This shell might conceivably burst in two ways—first, along a plane perpendicular to the axis, and second along a plane through the axis. In the first case it might break as shown at Fig. 2, *B* or *C*; in the second as shown at Fig. 3, *A*.

Taking the thickness, diameter and pressure the same as in Fig. 1, it is seen that the stress causing the tank to burst in the manner shown at *B* or *C* of Fig. 2 is 6,700 lb. per sq. in., exactly as in Fig. 1.

The stress causing the shell to burst as shown at *A*, Fig. 3, will likewise be 6,700 lb. at the curved ends, but greater along the straight portion. The latter, being the weakest portion, is therefore the place to figure. Consider a circular "hoop" one inch wide as marked at *A* and shown separately at *B*. The force tending to push the two halves apart is 700 lb. times the area of a rectangular strip 1 × 24 in., a total of 16,800 lb. Half of this, or 8,400 lb., is carried by each side; that is, by a strip of metal 1 × $\frac{5}{8}$ in. having $\frac{5}{8}$ sq. in. of cross-sectional area. The unit stress tending to cause this kind of a rupture is then 8,400 ÷ $\frac{5}{8}$ = **13,400** lb. per sq.in. This is twice the unit stress tending to produce the kind of rupture shown in *B* and *C* of Fig. 2.

For any other dimensions and fluid pressures the relative results would have been the same. That is, a seamless cylindrical tank with hemispherical ends will always burst along a longitudinal (lengthwise) plane, so only this case need be figured to find the factor of safety.

As ordinarily made, such tanks have both longitudinal and circular joints. As the efficiency of the weakest circular joint could be reduced to half that of the weakest longitudinal joint before the tank would be as weak one way as the other, it is generally sufficient to check the strength of the longitudinal joint only. Suppose there is one longitudinal double-riveted butt joint with an efficiency of 80 per cent. The solid metal along a longitudinal line has a factor of safety of 55,000 ÷ 13,400 = 4.1. The joint would have a factor 80 per cent of this, or 3.28. Since this is too low for safety with a tank containing steam, air or gas, it would be necessary to reduce the pressure considerably or build a stronger shell.

All that has been said applies only to shells that are relatively thin in proportion to the internal diameter.

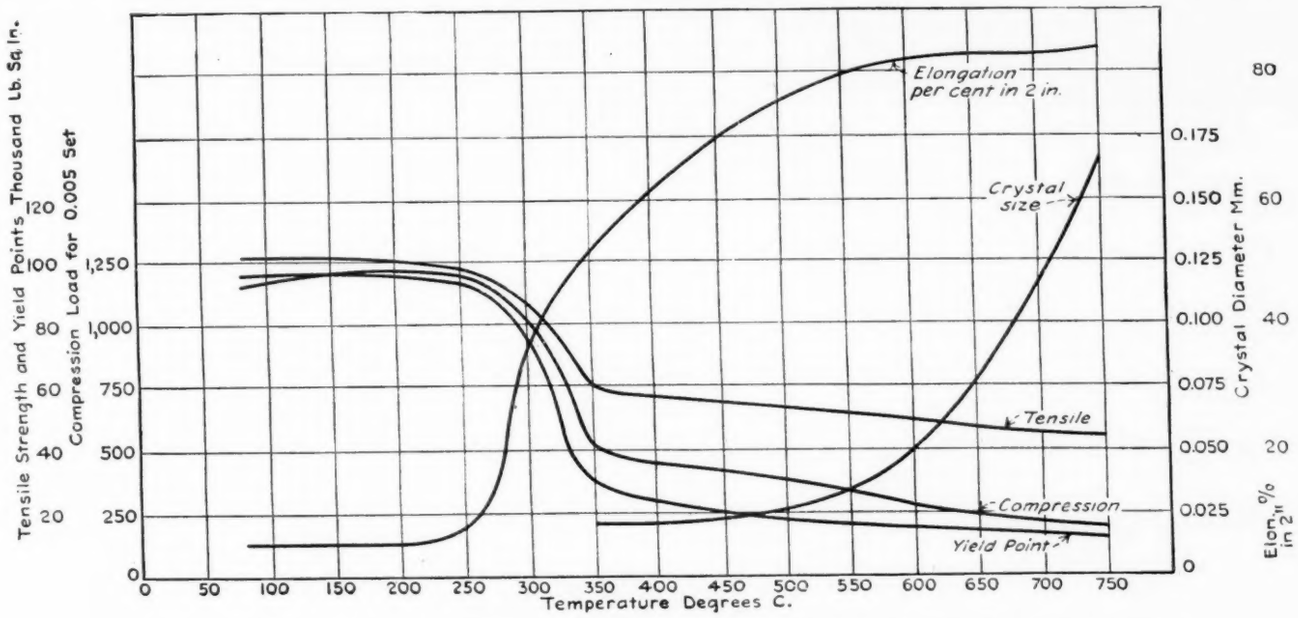
The Manufacture and Service Troubles of Condenser Tubes

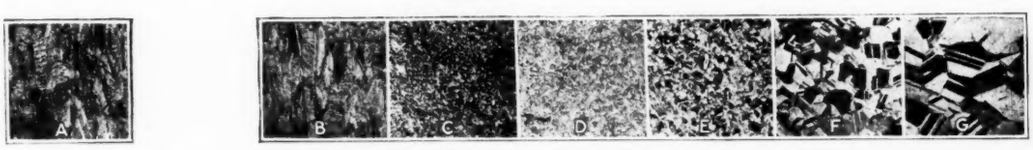
Condenser-tube troubles appear so diversified and conflicting, that no definite cause is satisfactorily indicated as responsible for the large increase of these difficulties during recent years. Circulating water has become more corrosive in character. Air bubbles, as well as pockets caused by water flowing around sharp corners, have recently been observed as responsible for corrosion.

IN A lecture before the Metropolitan Section of the American Society of Mechanical Engineers entitled "Condenser Tubes," delivered by W. R. Webster, vice-president of the Bridgeport Brass Co., on Nov. 8, the fact that better tubes are being made now than in years past, and yet the life in service is much shorter than formerly, was brought out. There are two general characters of failures in condenser tubes, one of which is the spontaneous longitudinal splitting of the tube, known as season cracking. The other is the corrosive action on the tubes of the cooling water and substances carried in with it. The former trouble is now of relatively minor importance, but because of the fact that, when encountered, it produces sudden and extensive leakage, the immediate result may be of greater consequence than failures due to corrosion, although the latter is gradual in approach and manifestation.

Owing to the erratic nature of corrosion and the large element of time involved, it cannot be said that sufficient facts have been established on the basis of scientific procedure to make it at this time possible for anyone, within very broad limits, to predict with certainty that any particular lot of tubes will endure under average service conditions in a measurably superior degree from other tubes differing from them in material respects. Careful study of the problem in all its aspects is well worthy of the interest of good engineering practice.

Condenser tubes originally were made of a mixture of about 70 per cent copper and 30 per cent zinc. This was principally for facility of manufacture. Later, 1 per cent of tin was substituted for 1 per cent of the zinc, which made a tube more resistant to corrosion. This was adopted by the British Navy and known as admiralty mixture. This material is used in connection with salt water or highly corrosive fresh water.


Tensile Strength and Yield Points Thousand Lb. Sq. In.
Compression Load for 0.005 Set
Crystal Diameter Mm.
Elon., % in 2"
Elongation per cent in 2 in.
Crystal size
Tensile
Compression
Yield Point
Temperature Degrees C.
0 50 100 150 200 250 300 350 400 450 500 550 600 650 700 750
0 20 40 60 80 100 120
0 250 500 750 1,000 1,250
0 0.025 0.050 0.075 0.100 0.125 0.150 0.175
0 20 40 60 80



EFFECT OF ANNEALING ON ADMIRALTY METAL, AND INDICATIONS OF TUBE SERVICE TROUBLE
CRYSTAL STRUCTURES, ETCHED WITH NH_4OH AND H_2O_2, MAGNIFIED 75 DIAMETERS

	Figure References						
	A	B	C	D	E	F	G
Temperature, Centigrade, of one hour annealing	*	250	350	450	550	650	750
Average actual grain size, millimeters, at given temperature	...	...	0.020	0.030	0.037	0.080	0.170

*Hard drawn (without anneal) 50 per cent.

Corrosion by air bubbles is shown at H; plug type dezincification, at I, and season cracking at J.

Season cracking in Admiralty tubes has largely been controlled by a moderate degree of annealing. Specifications requiring annealing were adopted by the United States Navy on the writer's recommendation about twenty years ago. Admiralty tubes cannot be manufactured by the piercing method on account of the physical quality of the metal. It is usual practice to make the initial casting a hollow cylinder about 4 in. in diameter, ⅝ in. thick by 50 in. long, representing the highest expression of the art of casting non-ferrous metal. After cleaning the casting, a mandrel is inserted in the bore, which is slightly larger than the inside diameter of the shell. This and the arbor are forced through a circular die ¼ in. smaller than the outside diameter. This reduces the cross-sectional area accordingly and elongates the casting 35 per cent. Inferior metal will not stand up under this treatment. Two more arbor draws are then made, after which drawings without arbor, and annealing processes follow each other until the finished diameter is reached.

Final Annealing Leaves Strength To Resist Handling

The final annealing is so regulated that it will remove the liability of season cracking, yet leave the tube hard enough to withstand handling. When a tube is drawn, the crystals are elongated. If the tube is subjected to an increasing temperature such as by an annealing, a point is reached where the crystals begin to break up. This action continues with rise of temperature until the original strained structure has disappeared and the mass consists of fine crystals free from indication of strain. As the temperature is raised more, the crystals increase in size progressively until the melting point is reached. The figure indicates the relative size of the crystals under different degrees of annealing. If the temperature of annealing before the draw is very high, the crystals will be large, and after the draw and subsequent anneal they will be larger in size than if annealed at a lower temperature in the beginning. In order to reach a size of crystal which is designated as fine and regular, it is necessary to employ low-temperature annealing, which does not cause the crystals to grow excessively. Heavy reductions are required in order to assist in breaking up the crystals by elongation so that subsequent annealing will make a small crystalline structure.

Curves in the figure illustrate the physical properties of Admiralty metal under the influence of temperature changes. The crystalline structures, corresponding to regular increments of temperature, are illustrated herewith.

Tube Failures and Crystalline Structure

Tube failures in a condenser are usually such that the difference in crystalline structure is not indicated as a clue to the trouble. Comparative examinations of large quantities of tubes both failed and unfailed show great variations in crystal size without a uniform record of either failure or success. Dr. C. B. Bengough advises: "In my experience it is very rare for tubes to fail from this cause," which refers to mechanical imperfections.

Air bubbles have been noted as being exceedingly active in causing corrosion. The characteristic of this failure is a bright surface along the inside of the tube as indicated at *H* in the figure.

Plug type dezincification is indicated by circular areas of small size eaten into the tube. This is shown at *I* in the figure. In this case the zinc appears to have been dissolved, leaving a spongy mass of copper behind, which tends to wash out. This proceeds at a rapid rate when once begun. Perforations may take place within a few months or even weeks from the time the tube is placed in service. This appears to be due to qualities of the condensing water, and in some cases condensers are installed with a certain amount of inclination so there will be complete drainage of the tubes when shut down.

Failures in which the wall of the tube is dissolved in an irregular manner, frequently in the region of inlet areas, have been proved to be due to the presence of air in the condensing water. This leaves a bright surface. Soaking the tubes in quiescent sea water at 40 deg. C. for about a week forms a protective coating, in many cases of much benefit.

Among preventive measures are recommended frequent cleaning of tubes and also the use of an iron protective plate properly attached to the tube sheet. The omission of the final pickling operation in the manufacture of tubing appears worthy of trial, as this will leave the outside scale, as a protective coating.

Manufacture by the cupping process, in which a solid bar is cast, rolled, pressed into a cup shape, and finally drawn into tubes, was described briefly.

The writer has for years vigorously opposed the contentions of those who hold the belief that crystalline structure or mechanical imperfections are causative accompaniments of corrosive attack, and in support of his position has presented evidence and cited authority. The attitude is not, and was not, dictated by any lack of appreciation of the importance of these factors from the viewpoint of good workmanship and practice, but results solely from the conviction that as long as their suspected presence is assumed to account for premature failures, the sufficient remedy for which lies in increased severity of inspection, no progress in determining and controlling the factors depending on condenser design and operation is to be expected.

Causes of Trouble Not Definitely Indicated

In the discussion that followed, it was brought out that during twenty years there had been no very definite conclusions produced in regard to the causes of condenser-tube failures. Evidence is of a conflicting nature. In one case where a 40,000-sq.ft. condenser had been retubed, a certain number were rejected. Of these, however, two hundred were nevertheless inserted in the condenser for test purposes. After an operating period of about two years it was found that about 40 per cent of the original tubes showed failures, as against 10 per cent of the rejected tubes on test. This indicated that careful inspection was not necessarily a guide as to the future life of the tubes. At first the life of a tube was two years. After regular, frequent cleaning intervals had been instituted, the life was increased to three years. No tubes have lasted four years. Draining a condenser was tried, but did not show a longer life. Dezincification, which was the original cause of failure, is now only a small factor. It appeared as if foreign elements, entering with the water, such as coal dust or ashes, acted as electrodes of a battery when they lodge in a condenser tube. Corrosion was principally at the top.

E. B. Ricketts brought out the fact that air bubbles had been especially serious, producing a deterioration that softened the metal so that it could often be cut with a knife.

The conflicting character of tube corrosion was strongly emphasized by Willis Lawrence. In one installation two condensers served the same turbine. In one condenser, tubes in the upper part corroded badly, and in the second, tubes in the lower part were corroded, although both represented the same manufacturer. In one case forty tubes were plugged, and in the second about seven hundred, during the same period of service. It had happened in one condenser that bad tubes were well distributed and in another case that there were a large number bad in one pocket. After twenty-one years of operation the causes are hazy in character.

Further discussion emphasized that the water compartments were not designed for good stream flow. Water, in turning sharp corners, tends to form air pockets and under such circumstances corrosion is inclined to be rapid. In some cases air may be injected which often stays for a considerable period in one location in the tube before being absorbed by the circulating water. Dezincification is the principal trouble.

Mr. Price, of the Scovil Mfg. Co., advised that the Bureau of Mines considered that grain crystal sizes were vitally connected with tube corrosion. Fine grain showed considerably less corrosion than coarse grain. In low-temperature annealing, such as 260 C. for one hour, there was no evidence of recrystallization, yet the strains were relieved.

Pinchot Tells of Huge Dividends on Coal

The American Academy of Political and Social Science Hold Conference at Philadelphia on the Price of Coal

AT THE conference of the American Academy of Political and Social Science held at Philadelphia, Nov. 16-17, "The Price of Coal, Anthracite and Bituminous," was considered at great length by the many authorities who addressed the conference.

Governor Pinchot, besides outlining his connection with the settlement of the anthracite strike, spoke at length on "Margins and the Consumer's Dollar" partly as follows:

No labor or other operating costs are paid from these margins. Out of each dollar realized by thirteen large anthracite operators from the sale of coal in the four pre-war years from 1913 to 1916, 61.3 cents went for labor, 12.3 cents went for supplies, 13.4 cents for general expenses, and 15 cents for the operator's margin. In the first three months of 1923, on the same accounting basis, 59.5 cents out of each dollar received by the operators went for labor, 10.8 cents for supplies, 11.5 cents for general expenses, and 18.2 cents for the operator's margin. These figures mean that the operators early in 1923 were adding to the price of their coal all their increased costs of every sort, plus 5.2 cents more out of every dollar. Now they are doing the same thing over again, by raising their prices more than their costs are raised. Under this system wage increases pay handsome dividends to the operators.

These figures cannot be dismissed with the statement that the money investment has greatly increased since 1913, for in fact it has not.

All the anthracite there is in America lies in Pennsylvania. In 1921 forty-eight companies produced 93 per cent of all the commercial tonnage, and seventy-four companies produced 99 per cent. Today ten railroad companies produce about three-fourths of it. It is commonly believed that financial relations exist between these ten railroad companies and some of the so-called independent operators. Common report asserts that there has long been and still is a common financial arrangement between these ten railroad companies and the anthracite carrying railroads.

It is commonly believed that most of these large companies make exorbitant profits through wholesale companies of their own, and that the wholesale companies affiliated with some of the large operating companies take an extortionate return for their services. Thus the Delaware, Lackawanna & Western Coal Co., the sales agent of the Glen Alden Coal Co., has paid dividends averaging 60 per cent for each of the six years from 1917 to 1922 inclusive.

It is also generally believed that many of the larger operators have common financial interests with many of the retailers.

In my opinion the whole combination is a hard-boiled monopoly whose prime interest in the public is that it shall consume their coal at their price. I share the common belief that the financial interests which so largely control anthracite from the mine to the coal bin help themselves to abundant profits as operators, take more profits as wholesalers, take still more in transportation, and in some cases take all they can get as retailers beside.

He further said in regard to dividends:

That these margins represent very large profits is shown by the size of the dividends paid by some of the companies. Thus the Lehigh Valley Coal Co., according to its own statement printed in "Poor's Manual," paid dividends of about 25 per cent in 1917, 21 per cent in 1918, 8.5 per cent in 1919, and 134 per cent in 1920. These were the dividends before the extra-large margins of 1921 and 1923. In 1921 this company declared a dividend of 72 per cent. Coke Brothers & Co. in 1920 paid a dividend of 91.7 per cent, and this was before the high margins of recent months. In the three years previous to 1920 this company paid dividends of 9 per cent, 10 per cent, and 24.5 per cent respectively. In 1921 its dividend was 83 per cent.

The Lehigh & Wilkes-Barre Coal Co. paid dividends of 13 per cent each year from 1909 to 1920 and in 1921 cut a melon of 226.5 per cent. In 1922 the year of scant operation because of the strike, this company paid a dividend of 59 per cent in 1920, 137 per cent in 1921 and 168 per cent in 1922. The Hillside Coal & Iron Company paid 70 per cent in 1920, 205 per cent in 1921 and 190 per cent in 1922.

Morris Llewellyn Cooke, director of the Pennsylvania Giant Power Survey, outlined the purpose of the Survey as follows:

Giant Power signifies not only a mammoth scale but a widely integrated electrical service and one benefiting largely from certain fundamental economies such as byproduct recovery at present either not practiced at all or only at isolated points. Giant Power also implies definite social ends, such as furnishing current generally to the small user, including both the farmer and the manufacturer.

Governor Pinchot and the 1923 Legislature which under him authorized the appointment of a Giant Power Survey Board hope through its activities and reports to make a substantial contribution to the conservation of national resources. For while we contend that electrical technology has reached the point where electrical power can be brought into almost universal use, we just as insistently demand that the easily preventable wastes resulting from our present methods of generating and distribution electrical current shall cease.

Will Conserve Coal Deposits

Pennsylvania is primarily interested in this new phase of the conservation movement because of her desire to safeguard her best coal deposits. For while she is supplying today approximately 40 per cent of the coal required for the nation's use, her deposits are far from inexhaustible. In fact the end is in sight, well within 100 years unless we can put a stop to some of the more flagrant and unnecessary wastes in the use of coal for power purposes. Other coal-producing states, such as West Virginia and Illinois, have relatively even more to gain because of the larger deposits yet remaining in the ground. In recent years we have come to appreciate the economy of water-power developments. It is not yet understood, however, that few water powers can be effectively utilized except as adjuncts to coal-burning power stations. Therefore with each step in the integration of electrical facilities a further inducement to water-power development is afforded, a further conservation of natural resources made possible.

In order to visualize what may prove to be the influence of Giant Power on coal let me suggest quite tentatively a few of the possible planks in the Giant Power program, without at this time pausing to present such supporting data as we may have: Large-sized stations—say not smaller than 500,000 kw., located at or near the mines and supplying current to a trunk system of transmission lines reaching 220,000 volts with large capacity to carry power long distances; byproduct recovery in some form; an integrated transmission and distribution system; the furnishing of current to the rural population; trunk-line railroad electrification; radical reductions in rates, especially to the small consumer and at least in proportion to their reduction in costs.

Anne Bezanson, Wharton School of Finance and Commerce talked on "Earnings Among Coal Miners"; Horace B. Drury of the Institute of Economics spoke on "Wages in the Coal Industry Compared with Wages in Other Industries"; John B. Andrews, American Association of Labor Legislation, "Needless Hazards in the Coal Industry."

At the session devoted to "Efficiency and Costs in Producing and Distributing Coal," the following program was carried out: "Historical Development of the Anthracite Industry," Francis Walker, Federal Trade Commision; "Where the Consumer's Dollar Goes," Samuel W. Tator, of the U. S. Coal Commission; "Costs in Distribution," by John W. Adams, U. S. Coal Commission; "Costs of Production, Including Investment and Profits of Operators," by David L. Wing; "Railway Rates on Bituminous and Anthracite Coal," by Charles Reitell, University of Pittsburgh; "Storage of Coal," by Ozni Porter Hood, U. S. Bureau of Mines.

On Friday evening there were addresses by Thomas Kennedy, United Mine Workers of America; S. D. Warriner, president of the Lehigh Coal & Navigation Co. and Gifford Pinchot, Governor of Pennsylvania.

On Saturday morning "Labor Relations in Coal Mining" was the topic, with addresses by F. G. Tryon, U. S. Geological Survey, on "The Effect of Competitive Conditions on

Labor Relations in Coal Mining"; J. H. Willits, University of Pennsylvania, "Labor Relations in Coal Mining"; Willard E. Hotchkiss, Federation of Clothing Manufacturers, "The Importance of Leadership in Labor Relations"; Rice Miller, Illinois Coal Operators' Association, "Some Problems in Labor Relations in the Coal Industry"; Francis J. Drum, United Mine Workers of America, "Labor Relations in the Coal Industry from the Union Standpoint"; General discussion, C. E. Lesher, Editor, *Coal Age*.

On Saturday afternoon Fuel Economy was the topic, with addresses by E. A. Holbrook, School of Mines, State College, on "The World's Present Fuel Resources," and Hugh Archbald, of the Victoria Mining Co., under the topic of "Efficiency and Cost Reduction in Coal Mining," gave a clear explanation of the relation between speculation and manner of operation of a coal mine:

It is necessary to realize that the first way of mining is massed craft work. The individual miner at the face digs, cleans and loads the coal. It is a finished product when it leaves him and all that is added to it is transportation out of the mine and over the railroad. That is the most efficient method for speculative operation as it involves the least investment. A third of the soft coal, 175,000,000 a year, is still mined in this fashion.

As soon as undercutting machinery is introduced into any mine, a break is made away from the most speculatively efficient condition and real production problems begin to enter. It is here that mine operation has had no leadership from such organizations as the American Institute of Mining Engineers.

For with undercutting machines to do part of the work of a miner, the most skillful part, there comes a division of labor. Instead of having two parts to the underground work—transportation and craft work—you have three—transportation, undercutting and loading. With divided labor there is a need of increasing the amount of supervision of the work by at least 50 per cent, considering only the division of labor.

Many a soft-coal operator has introduced machinery into his mine in the high hope of seeing his costs reduced and found instead, to his great discouragement, that his costs went up. The reason is simple. Intermittent operation is constantly dragging him back toward speculative operation, and he has changed his situation for being speculatively efficient. Machinery brings along with it the need of changing the layout of the work so that not alone men can work, but machinery can work efficiently. The openings which are driven must have different proportions. With craft work these proportions do not matter. With machines they are vital.

The consequence is that with machines not alone is there a need of increasing the quantity of the supervision, but there is also a need of increasing the quality. It is part of the production problem. It is hard to solve while intermittent operation is constantly forcing the employment of the cheapest and fewest number of bosses that idle days may not be expensive. For idle days are the largest single expense item in producing coal.

There was general discussion, then R. H. Fernald, of the University of Pennsylvania, spoke on "Fuel Wastes; Causes, Effects, Remedies."

"Steps on the Way Out" was the topic for the Saturday evening session, with addresses by Edward Eyre Hunt, of the U. S. Coal Commission, on "Summary of Recommendations of the United States Coal Commission"; William Draper Lewis, University of Pennsylvania, on "Legal and Economic Questions Involved in the Public Regulation of the Price of Coal"; F. R. Wadleigh, on "Substitutes for Anthracite as a Fuel"; William S. Kenyon, "A Code of Industrial Law."

Powdered Fuel Attracts New York A.S.M.E. Audience

The drawing power of the subject of powdered coal was demonstrated on the evening of Nov. 20, when approximately 500 members of the Metropolitan Section of the A.S.M.E. assembled to hear lectures on the subject by H. D. Savage, of the Combustion Engineering Corp., and F. A. Scheffler, of the Fuller Engineering Company.

Mr. Scheffler, who was the first to speak, told of the great activities in the pulverized-coal field. During the year 1922, fifteen million tons of coal was pulverized in the United States and Canada. The figures for 1923 will be about seventeen and a half million tons. Mr. Scheffler said that powdered-fuel furnaces were now designed on the basis of 7 to 10 ft. per sec. gas velocity. He understood that a special high-temperature firebrick was soon to be placed on the market at a moderate price. The use of such a brick would permit reducing furnace volume 25 to 35 per cent. The use of powdered coal is as safe as stoker firing, he claimed, if the equipment is properly designed. The distributing systems had been so far perfected that powdered coal could be forced through a pipe line one mile long. He pointed out that in making cost comparisons between stokers and powdered coal, the method of figuring often made the cost differential appear much higher than it really was. For example, there are preparation and handling costs with stokers as well as with pulverized-fuel apparatus. He gave figures showing that the operating costs with pulverized coal ranged from 57c. per ton for small stations to 43c. for those of medium size. The corresponding figures for stoker-fired plants were 38 and 36c., leaving the extra cost of pulverizing 19 to 7c. per ton. These figures do not apply to large central stations. The gain in efficiency due to pulverized coal was given as 10 per cent for large plants and 15 per cent for small.

In his paper on the same subject Mr. Savage stated that powdered fuel was much better established than many other innovations, such as high pressure, air heaters, bleeder heating, etc. Since January, 1923, powdered-coal installations for 1,300,000 sq.ft. of boiler-heating surface had been contracted for. These were for all kinds of coal under all kinds of conditions.

Among the advantages claimed for powdered coal were the ability to change the proportion instantly to meet changes in the heating value of the coal; complete oxidation of the sulphur, thus using its heat and preventing clinkers; ability to burn high-ash fuel with efficiency; ability to burn coal otherwise unavailable, thus saving on first cost and on freight.

As to first cost of the plant it was pointed out that it was unfair to make the comparison on the same total boiler-heating surface. The use of powdered coal permitted a considerable reduction in the boiler-heating surface for the same plant capacity. If proper consideration was given to such factors and comparison made between two plants, one independently designed for stokers and the other for powdered coal, the latter would show little, if any, greater first cost. The operating costs, according to Mr. Savage, were no higher for pulverized coal. A slide was shown giving the operating costs at Lakeside station, Milwaukee, for the first nine months of 1923. This will be reproduced in an early issue. Mr. Savage said the importance of the ash question had been greatly exaggerated. There were never any complaints, he said, from the neighbors. If the ash was there, it was too small to see.

In the discussion, directed by John H. Lawrence, chairman, Colonel Junkersfeld, of McClellan & Junkersfeld, said that the present Cahokia installation of eight 18,000-sq.ft. boilers would shortly be doubled and that his firm had seen fit to recommend pulverized fuel for many other installations. Mr. Keeth, of the same company, outlined briefly the problems incident to the recent starting up of the Cahokia plant. The problems encountered, he said, were comparable with those in an ordinary stoker-fired plant. The maximum rating carried so far is 235 per cent, no attempt having been made to force the boilers. The coal used has 5 per cent moisture, 13 per cent ash and 11,500 B.t.u. The flue temperature is around 500 deg. At present 6-ton mills are used, but it is planned to increase these to 15-ton. There has been no trouble with slag.

A member of the J. G. White Engineering Corporation told of satisfactory experiences with a powdered-coal installation erected in Colombia and operated with native labor.

Professor Christie, of Johns Hopkins University, spoke of the possible incomplete combustion in the outer envelope of the gas due to overcooled furnace walls. The answer

to this problem, he said, could be found in preheating the air and in imitating the bunsen burner. In this way high efficiency could be obtained without destruction of refractories. As to the ash he thought there must be some economic use for such a finely divided material.

Mr. Frost, of the (New Jersey) Public Service Co., stated that his company had decided in favor of stokers rather than pulverized coal.

R. J. S. Pigott, of Stevens & Wood, said that heating surface in water screens was almost as effective as that in the bottom row of boiler tubes, and that it might evaporate from 40 to 80 lb. of water per hour per square foot. Except for a foot or two of the outer envelope, he thought that the wall temperature had little effect on the flame temperature. He agreed with Professor Christie that the thing to do was to preheat the air.

Commenting on flame temperature, George Orrok said that it was impossible to reduce the temperature of the body of the flame by surrounding it with cold walls. This could be seen by lighting a candle in a cold boiler setting. The flame temperature would remain as usual. The same holds for an oil or powdered-coal flame, which would run from 3,200 to 3,500 deg. regardless of the furnace-wall temperature.

Another speaker related some unfortunate experiences in attempting to fire Scotch marine boilers with powdered coal. The boilers had furnaces 4 ft. in diameter. The efficiency obtained ranged down to 40 per cent. The speaker stated, however, that he thought the poor performance due to insufficient furnace volume rather than to powdered coal itself.

Pulverized Coal, Equipment and Operation

Pulverization renders available coals so strong in sulphur that they are unusable and unmarketable, as well as those with ash which fuses at so low a temperature as to render other means of combustion unworkable. Similarly, it removes the difficulties of burning lignite so that those vast deposits of half-formed coal found in the western part of North America yield to it as well as true coal. These and other facts were well brought out in a paper entitled "Pulverized Coal," by F. W. Dean, which was read before the National Association of Cotton Manufacturers at their semi-annual meeting in Boston, Mass., Oct. 31, 1923.

The reasons for economy of coal by pulverization may be stated as follows:

1. Practically all of the coal is burnt, there being almost none in the ash.
2. The combustible elements of the coal are more completely burnt to CO_2 than by any other means
3. All the coal in starting is utilized.
4. When stopping, no coal is left in the furnace.
5. No banking coal is used, and no stand-by losses occur except by radiation.
6. Coal that fails to burn in suspension and falls to the bottom of the furnace burns slowly.
7. Accurate control of air to accompany the fuel used.
8. Instant change of rate of firing to meet needs.
9. Low draft requirements.
10. Low labor costs.

There is ordinarily less than ½ per cent of carbon in the ash against 15 to 40 per cent when burned by other means. Of more importance is the high percentage of CO_2 that can be realized by this method. A pound of coal burned to CO_2 yields about 14,500 B.tu., and when burned only to CO it gives 4,300 B.t.u., or approximately one-third as much heat. Less surplus air is necessary than by other methods, since a more complete mixing of the air and coal is attained with pulverized fuel. The capacity of a given chimney is greater when pulverized fuel is used.

Generally speaking upward of 70 per cent of the pulverized product should pass through a screen having 40,000 openings per square inch. This means that in a screen formed by 0.002-in. diameter wire, with 200 spaces between the wires in one inch of mesh, with two sets of wires at right angles, there are 40,000 openings for the material to pass through. Coal of this character burns like gas, giving a very high temperature, especially as it is necessary to have little excess air.

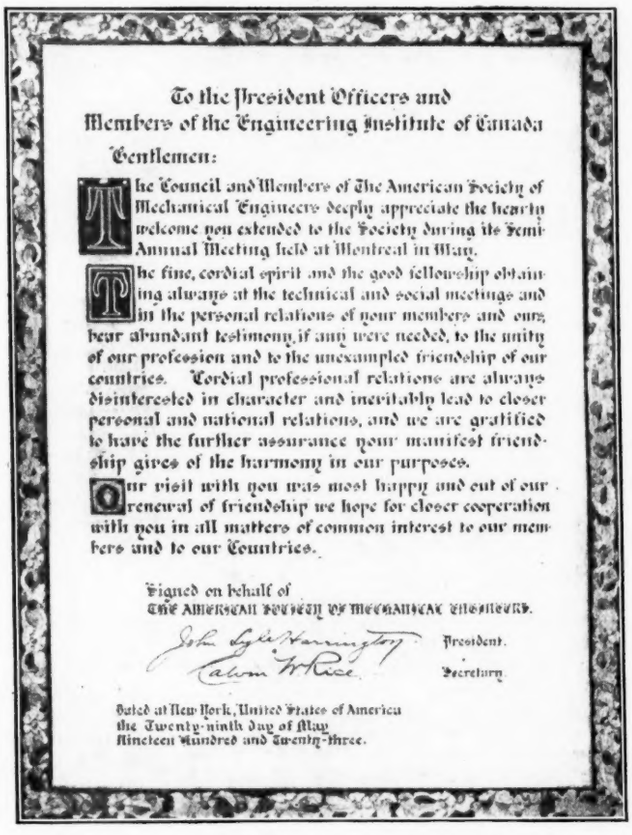
Fire is ordinarily started by means of igniting wood, oily waste or a stream of oil at the beginning. When shut down, there are no losses equivalent to banked fires as in other means of combustion

Slag is troublesome on account of the high temperatures. The usual method of preventing melting of the ash on the brickwork is to provide refractories that contain air spaces, thus supplying air from the outside and delivering it to advantageous points in the furnace. The air circulation cools and helps to maintain the brickwork, and the ash falls as a fine light-colored powder. Another means of accomplishing the same result is to cool the bricks by means of water circulation. Cooling coils of this kind add to the effective heating surface of the boiler. Sometimes both methods are used in combination.

Since the fuel is in small particles and is fed with high velocity the furnace requires a long travel for complete combustion of the flame. A volume of 2 cu.ft per horsepower to be developed is the usual allowance. Volumes should not be determined from nominal ratings.

There is little or no smoke from pulverized-fuel furnaces. It is best to use balanced draft in order to utilize the furnace volume to the best advantage. Scarcely any dust is noticeable near a pulverized-fuel installation, and that which falls is fine, clean, sootless and much less harmful than dust emitted from hand- or stoker-fired plants. It is sometimes necessary to dry coal. This depends on the amount of moisture in the coal as well as the means employed for pulverizing and treating.

There are two general methods of pulverizing coal—the impact and the grinding methods. On the former coal is splintered by rapidly revolving paddles. This is on the principle of hitting a piece of coal with a baseball bat and cracking it. The resulting fine dust is drawn out of the pulverizer by means of a fan and blown into the furnace. The second method is that of grinding or hammering, where coal is crushed between moving and stationary pieces that might be described as hammer and anvil elements.



To the President Officers and
Members of the Engineering Institute of Canada

Gentlemen:

The Council and Members of The American Society of Mechanical Engineers deeply appreciate the hearty welcome you extended to the Society during its Semi-Annual Meeting held at Montreal in May.

The fine, cordial spirit and the good fellowship obtaining always at the technical and social meetings and in the personal relations of your members and ours, bear abundant testimony, if any were needed, to the unity of our profession and to the unexampled friendship of our countries. Cordial professional relations are always disinterested in character and inevitably lead to closer personal and national relations, and we are gratified to have the further assurance your manifest friendship gives of the harmony in our purposes.

Our visit with you was most happy and out of our renewal of friendship we hope for closer cooperation with you in all matters of common interest to our members and to our Countries.

Signed on behalf of
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

John Lyle Harrington, President.

Calvin W. Rice, Secretary.

Dated at New York, United States of America
the Twenty-ninth Day of May
Nineteen Hundred and Twenty-three.

FAC-SIMILE OF A LETTER OF APPRECIATION SENT BY A.S.M.E. TO THE ENGINEERING INSTITUTE OF CANADA

News in the Field of Power

Canal Revenues Increased by California Oil

Owing to the development of California oil and its movement to the East Coast, the operations of the Panama Canal and auxiliary agencies during the fiscal year ended June 30, 1923, were record breaking in volume both as to financial returns to the United States government and aid rendered to the shipping world. The annual report of Governor Morrow, made public Nov. 18, shows that the canal netted a profit of $12,063,880 from all sources excluding the Panama Railroad Steamship Line, as compared with $9,201,613 in the fiscal year 1922. Traffic operations increased on a scale that has brought the waterway to a point rapidly overhauling the Suez Canal.

The net income from tolls and other miscellaneous receipts grouped under the head of transit revenue was $10,001,066, as compared with $3,466,674 in 1922. The net profits from auxiliary operations totaled $1,140,642, as against $323,259, while those conducted by the Panama Railroad Co., excluding the steamship operations, showed a profit of $922,171 against a loss of $710,301 in 1922. The report also showed that the number of commercial transits increased from 2,736, to 3,967, net tonnage from 11,417,459 to 18,605,786, tolls from $11,107,832 to $17,508,199 and cargo tonnage from 10,884,910 to 19,567,875.

EXPLOSION OF BOILER AT GREENVILLE, ALA.

Inspection Might Have Prevented This Explosion

On Oct. 27 a horizontal tubular double-lap-seam boiler exploded at the Hartlet Gin, Greenville, Ala., causing the death of two men and the serious injuring of nine others. The explosion was heard a distance of twelve miles, steelwork and brickwork being hurled high in the air and striking adjoining buildings. Work cars on a side track were so badly damaged as to appear to have been in a wreck. One of the freaks of the accident was the escape from injury of a mule which was standing in front of the boiler room. Although covered with the wreckage, the mule shook his way to freedom and trotted off unhurt.

As is usual in case of boilers exploding from unknown causes, sudden pumping of water into the boiler after the water had disappeared from the water gage is assigned as the cause of the disaster.

While the age of the boiler is unknown, advices received from several sources indicate that it was not insured and had not been inspected by an inspector. Whatever the cause of this disaster, there is every reason to believe that an adequate inspection by a skilled and intelligent engineer would have enabled the defect to be discovered in time to prevent trouble.

Federal Power Commission Will Be Sued by New Jersey

The United States Supreme Court on Nov. 19 granted a motion permitting the State of New Jersey to file an original bill against the Federal Power Commission to enjoin the Commission from exercising authority over water-power development within New Jersey and to test the constitutionality of the Federal water-power act. The New Jersey case will be expedited and will be heard jointly with a similar suit against the Commission by the State of New York shortly after the first of the year.

Spring Strike in Soft-Coal Mines Predicted

According to L. A. Snead, president of two Washington coal companies and former assistant to the Federal Fuel Distributor during the War, "A coal strike in the union bituminous fields affecting about 75 per cent of the nation's soft-coal production is certain." "There are three major problems on which neither the operators nor the miners will arbitrate. One is the wage question, another is the check-off and the third is the national-versus-district agreements."

N.A.S.E. Announces Prizes for Coming Engineering Contest

The National Educational Committee of the National Association of Stationary Engineers has announced an "Engineering Problem Contest," in which five prizes ranging from $400 to $50 will be given. Beginning with the November, 1923, issue of the *National Engineer* and continuing until July, 1924, there will be printed in each issue a list of thirty questions on subjects of arithmetic, through mensuration, heat, pneumatics, hydro-mechanics, applied mechanics, pumps and condensers, steam boilers and steam engines. One subject to be treated each month. Those interested in entering the contest can obtain a copy of the rules and regulations from Frederick L. Ray, 1124 Victory Court, Anderson, Ind.

Central Maine Power Co. Plans Hydro-Electric Project

Plans for hydro-electric developments aggregating 110,000 hp. and costing about ten million dollars were announced during the past week by the Central Maine Power Co. To a representative of *Power* Walter S. Wyman stated that it is hoped to start construction during 1924. One development involves the building of a dam across the Kennebec River above Lewiston for the delivery of 50,000 hp., and the other includes a 90-ft. dam across the Kennebec River above Bingham for the development of 65,000 hp. The latter project includes a storage reservoir of 1,000,000,000 cu.ft. capacity. The estimates of cost include additional generating plant, transmission lines and substations. An enlargement of the Lewiston auxiliary steam plant of the company is

now under way, together with an interconnection between the Central Maine Power Co. and the Cumberland Power & Light Co., of Portland. F. H. Mason, Waterville, Me., is chief engineer of the Central Maine Company.

Power Conference Issues Invitations

Tentative lists of persons to be invited to participate in the World Power Conference have been prepared by the American Committee. The invitations are being limited to those who are identified with engineering problems and it is expected that about 300 to 500 invitations will be issued. Probably there will be only about 100 acceptances out of this number.

Adult Vocational Training Promised

The Federal Board for Vocational Education announces that adult wage earners who wish to receive technical training that will help them in their occupations, may now secure such training from the public schools.

Under the Federal and State Vocational Education Acts, funds are now available to assist the public schools in each community to set up adult evening classes for wage earners. To receive such instruction, it is necessary that a group of workers, employed in a common industry take the initiative by organizing themselves into a class. Next, they should present themselves to the local school authorities and request an instructor. The local school board is empowered to provide an instructor and a classroom to the applicants. Thereupon the class begins its session on a regular nightly schedule.

Standardization of Sizes of Anthracite Considered

The American Society for Testing Materials' committee, D-5, on coal and coke, is now taking up the subject of standardization of anthracite sizes. A meeting of Subcommittee IV, of Committee D-5, on standard sizes, was held in the laboratories of the United States Testing Co. on Oct. 19, which was attended by the members of the subcommittee, representatives of dealers in coal, consumers and combustion engineers. The work of the subcommittee involves the following:

1. Names and the sizes as used in the trade.
2. Method of sampling a shipment of anthracite and making the sieve tests.
3. Sieves to be used in test (given size shall pass through a specified opening and be retained on a specified opening.)
4. Tolerances of allowable undersize and oversize.
5. Specifications for the sieves to be used.

As a result of the meeting names and sizes as used in the trade were tentatively agreed upon.

Human Engineering Paramount

Charles M. Schwab Places Good of Humanity Above Material Achievement

Responding to the presentation of a certificate of honorary membership in the Philadelphia Engineers' Club at a banquet given in his honor at the Bellevue-Stratford last Tuesday evening, Mr. Schwab declared that the engineer has done more for advancing American industry than any other class. Moreover, he was proud that we are called a material nation because as such we have worked for the advancement of civilization and the good of mankind. Without detracting from the material progress, however, he considered human relations as paramount and believed the greatest engineers to be those who are directing their energies toward the problem of sound and amicable relations between capital and labor. He compared industry to a three-legged stool, the legs being represented respectively by capital, management and labor, the collapse of any one of which causes the whole structure to fall.

Mr. Schwab remarked that the division in industry as between labor and capital is not always equitable, but the efforts of all engaged in industry should be to make that division more equitable. This is the direction which human engineering is taking today and in which American engineers are leading.

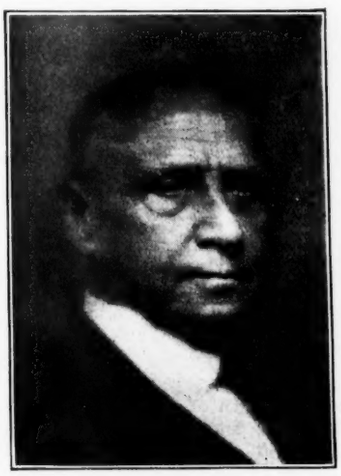
The speaker modestly disclaimed personal credit for many of his achievements as referred to by previous speakers and remarked that the credit belonged very largely to the engineers who had been associated with him in his various undertakings. The guests were surprised to hear Mr. Schwab remark that had he his life to live over again and a choice of vocation at his command, he would select that of teaching in preference to being an industrialist or financier.

Alba B. Johnson acted as toastmaster, and the other speakers were Prof. R. H. Fernald, president of the Engineers' Club, Mayor Moore of Philadelphia and Rear-Admiral A. H. Scales, U. S. N., who referred to the dependence of the Navy and the Merchant Marine upon the steel industry and the co-operation which they had received from Mr. Schwab during the World War. Julian Kennedy reminisced on the early days of the steel industry when he was associated with Mr. Schwab at the Edgar Thompson Steel Works. At the close of his remarks he pointed out that Mr. Schwab had become president of the Carnegie Steel Works at the age of thirty-five.

Obituary

Charles D. Noyes died on Nov. 19, at the age of seventy, at his home in Wellesley Hills, Mass. For twenty-eight years he was associated with the Hartford Steam Boiler Inspection & Insurance Co. and was widely known in steam-plant inspection and operating circles. He also had an extended railroad experience in early life. He was a native of Greenwood, Maine.

Lewis H. Nash, president of the Nash Engineering Co., South Norwalk, Conn., died Saturday, Nov. 17, after a brief illness. He was 71 years of age. Mr. Nash was born in South Norwalk, and after his early education, he served an apprenticeship at the Norwalk Iron Works, becoming an expert machinist, and afterward entered the Stevens Institute of Technology, where he pursued the full course of mechanical engineer, class of 1877, and his death is the first break in the ranks of graduates of that class. Upon graduation he received the degree of mechanical engineer, and in



LEWIS H. NASH

1921 the Institute conferred upon him the degree of doctor of engineering.

From 1877 to 1905 he was chief engineer of the National Meter Co., Brooklyn, N. Y., and was the inventor of the Nash Water Meter and Nash Gas Engine made by that company. In 1905 he organized the Nash Engineering Co., widely known as manufacturers of the Nash hydro-pneumatic turbine air compressors and vacuum pumps. Mr. Nash was prominent in business, civic and fraternal circles and during his lifetime secured a large number of patents on mechanical devices used in products of the National Meter Co. and the Nash Engineering Co.

New Publications

Diesel Engines. By L. H. Morrison, associate editor of *Power*. Published by McGraw-Hill Book Co., Inc., New York City. Cloth; 5¾x9 in.; 585 pages; 385 illustrations. Price, $5.

The book that finds a welcome place in the library of the operating engineer is one that tells the story in a practical and easily understandable manner and presents a large amount

of information and data that could be obtained only by a broad experience in that particular field. These requisites have been fully met by this volume. The subject matter has been compiled with a view to supplying the man who operates engines of this type, with the necessary facts to do so successfully and economically The early chapters cover a survey of the Diesel-engine industry in America and treat in a thorough manner the principles of operation, design and construction of all the engines of this type manufactured in this country. The general descriptions of the component parts are given in separate chapters that follow. Chapter V treats of the "Economic Status of the Diesel Engine" and covers rather exhaustively subjects that are of vital interest to all who own or operate this type of equipment; namely, first cost, fuel costs, labor costs and maintenance. Results of tests made on several engines under various conditions are also given. The chapters that follow cover oil-engine installation, Diesel-engine frames, cylinders and cylinder heads, pistons and piston pins, connecting rods, main bearings and crankshafts, admission and exhaust valves, fuel-injection valves, fuel pumps, governors, air-compression systems, Diesel fuel oil, lubrication, airless-injection oil engines, indicating the Diesel engine, operating procedure, testing of oil engines and marine rules. The subject matter in this volume, together with the many illustrations which it contains, represents a wealth of information for those responsible for the operation and upkeep of oil engines.

Hydro-Electric Power Stations. By David B. Rushmore and Eric A. Lof. Published by John Wiley & Sons, Inc., New York City, 1923. Cloth; 6x9 in.; 830 pages; 437 illustrations; 66 tables. Price, $7.50.

This is the second edition of this book, which has been revised and brought up to date so as to set forth recent practices in hydro-electric power development. The rapid advances made in recent years in water-power development have led to radical changes in both the hydraulic and electrical parts of the station. Draft-tube and runner designs have undergone the greatest changes, and the large sizes of the units have made necessary many changes in the design of switching and protective equipment. It is to set forth these new practices that this new edition has been published.

The book is divided into ten chapters and an appendix. Chapter I is a general introduction dealing with the history of water power and electrical development, water powers of the world, classification of developments, etc. Hydrology is the subject of Chapter II and treats of properties of water, rainfall, disposal of rainfall, stream flow and energy of flowing water. In Chapter III the various practices in dam and headworks construction are set forth, with numerous examples of the different types of construction. Water conductors, surge tanks, gates and valves are treated in Chapter IV, and in Chapter V storage reservoirs have been given attention. In Chapter VI numerous examples of modern hydro-electric power-house construction are given as well as a discussion on the feature of design and putting the station into service. In Chapter VII hydraulic equipment, turbines, draft tubes, governor pressure regulation and flow meter are the chief subjects considered. In Chapter VIII there are 400 pages devoted to electrical equipment in the plant, such as generators, exciters, relay protection, switching equipment, transformers, synchronous condensers, automatic regulation, etc. Chapter IX is devoted to the economic aspects of water-power development. In this chapter the cost figures for a number of installations are given, but since most of these are for conditions existing prior to 1917, they are of little value at present other than as a general guide. Chapter X goes into organization and operation of power plants. In the appendix is given a table of 160 of the principal power systems of the world with their voltages, frequencies, etc. This appendix also contains the text of the Federal Water-Power Act and Standard Testing Code for Hydraulic Turbines.

The authors for many years have been connected with one of the largest engineering organizations in the world, and have had an excellent opportunity to collect valuable information over a wide range of experience and find out what are the essentials in dealing with larger engineering problems. Formulas are given where necessary in various parts of the book, and problems are worked out. However, the text is not a mathematical treatise, but for the most part a practical presentation in readable language of the problem involved in the construction, management and operation of hydro-electric plants. Although the book is written primarily for the engineer and student of engineering, it contains a great deal of information of value to hydro-electric plant operating engineers and should find a wide field of usefulness among these classes of readers.

Coming Conventions

American Association for the Advancement of Science. Burton E. Livingston, Smithsonion Institution Bldg., Washington, D. C. Seventy-fifth anniversary meeting at University of Cincinnati, Cincinnati, Ohio, Dec. 27-Jan. 2.

American Engineering Council of the F.A.E.S. L. W. Wallace, 24 Jackson Place, Washington, D. C. Meeting at Washington, D. C., Jan. 10-11.

American Institute of Chemical Engineers. J. C. Olsen, Polytechnic Institute, Brooklyn, N. Y. Winter meeting at the New Willard Hotel, Washington, D. C., Dec. 5-8.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Midwinter convention at Philadelphia, Feb. 4-8.

American Society of Heating & Ventilating Engineers. C. W. Obert, 25 West 39th St., New York City. Annual meeting at New York City, Jan. 22-25.

American Society of Mechanical Engineers, Calvin W. Rice, 29 West 39th St., New York City. Annual meeting at New York City, Dec. 3-6.

American Society of Refrigerating Engineers, William H. Ross, 35 Warren St., New York City. Nineteenth Annual Convention at Hotel Astor, New York City, Dec. 3-5.

American Society of Safety Engineers. Genevieve S. Wood, 29 West 39th St., New York City. Annual meeting at New York City, Jan. 18.

Engineering Institute of Canada. Fraser S. Keith, 176 Mansfield St., Montreal, Canada. Annual General Meeting at Montreal, Quebec, Jan. 22—Ottawa, Ontario, Jan. 23-24.

Iowa Engineering Society. Blanche Veig, Acting Secretary, 406 Flynn Bldg., Des Moines, Iowa. Annual meeting at Cedar Rapids, Jan. 29-Feb. 1, 1924.

Kansas Engineering Society. C. V. Waddington, Kansas Gas & Electric Co., Wichita, Kans. Convention at Wichita, Kans., Dec. 11-13.

National Association of Practical Refrigerating Engineers; Ed. H. Fox, 5707 W. Lake St., Chicago. Fourteenth Annual Convention at Memphis, Dec. 12-16.

National Exposition of Power and Mechanical Engineering; Charles F. Roth, Room 1102, Grand Central Palace, New York City. Power Show, Dec. 3-8.

National Marine Engineers' Beneficial Association. Geo. A. Grubb, 313 Machinists' Bldg., Washington, D. C. Annual Convention at Franklin Square Hotel, Washington, D. C. Jan. 21-26.

Society of Automotive Engineers, Coker F. Clarkson, 29 West 39th St., New York City. Annual meeting at General Motors Bldg., Detroit, Mich., Jan. 22-25.

Society Affairs

The Atlanta Section of the A.S.M.E. will have as its speaker on Dec. 21, Nathaniel P. Pratt, who has as his subject, "An Ideal Engineering Education."

The Newcomen Society, South Kensington, S.W.7., London, will hear a peper, "The Dynamics of Leonardo da Vinci," by Ivor B. Hart, at its annual general meeting on Dec. 6.

The Hartford Section of the A.S.M.E. will hear George A. Orrok speak on "Progress in Power-Plant Development During the Past Few Years," on Dec. 10. The talk will be illustrated by slides.

The Baltimore Section of the A.S.M.E., jointly with the M.I.T. Alumni Association, will have as speaker at their meeting on Dec. 14, Dr. Stratton President of the M.I.T., whose topic will be "The Engineer."

The Lehigh Valley Section of the A.S.M.E. will hold a joint meeting on Nov. 30, with the Engineers' Club at Bethlehem, Pa. Prof. C. F. Hirshfeld, of the Detroit Edison Co., will speak on "Future Utilization of Coal."

The Franklin Institute of Pennsylvania will hear a lecture in the lecture room of the Institute at Philadelphia on Dec. 19, by W. F. G. Swain, Professor of Physics, University of Chicago, on "The Structure of the Atom."

Business Notes

The F. C. Richmond Machinery Co., representative for the Conveyor Corporation of Chicago, has moved its Salt Lake City offices and salesrooms to 320 West Second South St.

The United Machine & Manufacturing Co., of Canton, Ohio, announces that W. F. Pattengill, formerly secretary and treasurer of the Metropolitan Steel Co., has joined its organization as assistant to the president, and that H. S. Stockdale has been appointed district sales manager with offices in the Oliver Bldg., Pittsburgh, Pa.

The Chain Belt Co., Milwaukee, Wis., has established direct factory branches and warehouses in Portland and Seattle. Allen C. Sullivan has been placed in charge of the Northwest territory with headquarters at 67-69 First St., Portland, Ore.; Don B. Catton will be special representative for the Portland office; William F. Nichols will take care of the Seattle and British Columbia territory from the Seattle office, which is at 1040 Sixth Ave., South.

The General Electric Co. announces that the publication and advertising departments of its company will be combined on Dec. 1, with Martin P. Rice as manager. C. H. Lang will continue as assistant manager, and T. J. McManis, formerly with the Edison Lamp works at Harrison, N. J., will also be an assistant manager. An advertising council has been created, with B. G. Tremaine, J. R. Lovejoy, George F. Morrison, F. S. Terry and A. D. Page as ex-officio members. J. G. Berry will be chairman of the council, C. H. Lang, secretary, Bruce Barton, advertising council, and the other members will be M. P. Rice, T. J. McManis, P. B. Zimmerman, George C. Osbourne, L. P. Sawyer and G. P. Baldwin.

The Conveyors Corporation of America announces that Jay C. Lathrop, Cincinnati district representative, has removed his office from 326 W. Madison St., Chicago, to 503 Neave Building, Cincinnati, Ohio.

Trade Catalogs

Pulverizer, Coal — K-B Pulverizer Corp., 92 Lafayette St., New York City. Bulletin No. 20, "Pulverburner for Powered Coal Firing," describes with illustrations the operations of this mechanical unit for burning powdered fuel.

Engines, Vertical Steam — Clarage Fan Co., Kalamazoo, Mich. Catalog 61 describes with many illustrations these engines of high, medium, moderate and low pressures. Tables of weights, specifications, horsepower, and dimensions are given.

Vitreosil — The Thermal Syndicate, Ltd., 350 Madison Ave., New York City. Booklet "Vitreosil Data," gives the general characteristics of this fused quartz or silica which has insulating qualities, high temperature resistance, and high dielectric strength.

Potentiometer Pyrometers—Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia, Pa. Bulletin 872 gives descriptions of these measuring instruments and tells of the importance of flue-gas temperatures in efficient boiler operations. Illustrations are given which show the operation of the instruments.

Petroleum Inspectors Manual—The C. J. Tagliabue Manufacturing Co., Brooklyn, N. Y., has issued a revised edition of its manual for inspectors of petroleum. This covers the methods of making gravity, color, viscosity, flash point, carbon and some thirty other tests. While primarily for the use of oil inspectors, this book should be of value to engineers of plants purchasing large amounts of oils.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Nov. 12, 1923	Nov. 19, 1923
Pool 1,	New York	2.75@3.25	$3.25
Smokeless,	Columbus	2.10@2.65	2.35
Clearfield,	Boston	1.70@2.60	2.50
Somerset,	Boston	4.25@4.35	2.75
Kanawha,	Columbus	1.75@2.00	2.00
Hocking,	Columbus	1.75@2.00	2.00
Pittsburgh No. 8	Cleveland	1.85@1.95	1.95
Franklin, Ill.	Chicago	2.25@3.00	2.75
Central, Ill.	Chicago	2.00@3.25	2.25
Ind. 4th Vein,	Chicago	2.50@2.75	2.75
West Ky.,	Louisville	1.50@1.85	1.85
S. E. Ky.,	Louisville	1.50@2.25	2.25
Big Seam,	Birmingham	1.75@2.15	2.15

New York—Nov. 22, light oil, tank car lots, 28@34 deg. Baumé, 3⅝c. per gal., 36@40 deg. 3¾c. per gal., f.o.b. Bayonne, N. J.

St. Louis—Nov. 13, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.50 per bbl.; 26@28 deg., $1.60; 28@30 deg., $1.65; 30@36 deg., gas oil, 4@4⅛c. per gal.; 36@40 deg., distillate, 4⅜@4⅝c. per gal.

Pittsburgh—Nov. 16 f.o.b. local refinery, 30@34 deg., fuel oil, 5¼c. per gal.; 36@40 deg., fuel oil, 5⅜c.; 34 deg., neutral 8c. per gal.

Dallas—Nov. 17, f.o.b. local refinery, 26@30 deg., $1.35 per bbl.

Cincinnati—Nov. 22, tank-car lots, f.o.b. local refinery, 26@30 deg. Baumé, 5c. per gal.; 30@32 deg., 5¼c.; 38@42 deg., distillate, 6¼c. per gal.

Philadelphia—Nov. 16, 26@28 deg., $1.63@1.67½ per bbl.; 16@20 deg., $1.59 @1.63; 14@16 deg., $1.35@1.41½.

Chicago—Nov. 9, 24@26 deg., $1.72 per bbl.; gas oil, 32@36 deg., $1.92 per bbl.

Boston—Nov. 20, tank-car lots, f.o.b. heavy oil, 12@14 deg. Baumé, 4¼c. per gal.; light oil, 28@32 deg. Baumé, 6¼c. per gal.

New Plant Construction

PROPOSED WORK

Ark., Little Rock—The Arkansas Light & Power Co., Pieron St., H. C. Cough, Pres., plans improvements and extensions to cost several hundred thousand dollars.

Calif., Long Beach—The city plans an election to vote $3,500,000 bonds for the construction of school buildings and additions.

Calif., Los Angeles—A. C. Blumenthal, et al., Loew State Bldg., is having plans prepared for the construction of an 11 story, 125 x 145 ft. hotel on Wilshire Blvd. and New Hampshire St. Estimated cost $1,000,000. Walker & Eisen, 330 Pacific Finance Bldg., Los Angeles, Archts. Equipment detail not reported.

Calif., Los Angeles—A local syndicate is having plans prepared for the construction of a 14 story, 145 x 160 ft. office building on 9th and Hill Sts. Cost will exceed $1,000,000. Austin, Ashley & Farwell, Detwiler Bldg., Los Angeles, Archts. Equipment detail not reported.

Calif., Manteca—The South San Joaquin Irrigation Dist., G. W. Parker, Supt., will purchase two 1,000 g.p.m. and two 750 g.p.m. centrifugal pumping plants.

Calif., Oakland—The Pacific Gas & Electric Co., 445 Sutter St., San Francisco, is having plans prepared for the construction of a 2 story substation on 21st St. near Grove St., here. Estimated cost $280,000. Private plans. Electrical equipment has been purchased.

Conn., Hartford — The Bd. of Contract and Supply, Municipal Bldg., will receive bids until Dec. 4 (change of date), for the construction of a 3 story school building on Maple Ave. Estimated cost $1,200,000. E. T. Wiley, 75 Pratt St., Hartford, Archt. Equipment detail not reported. Noted Nov. 13.

Fla., Tavares—J. M. Gillespie, P. O. Box 128, is in the market for one electric pump for deep well and one for shallow well; one windmill; 40 ft. steel tower and a 400 gal. steel tank.

Ill., Shelbyville — The Burke Coal Co., 1626 Chemical Bldg., St. Louis, J. T. Burke, Purch. Agt., is in the market for an electrically driven coal conveyor, tipple machinery, pumps, motors, etc., for mine, 500 ton daily capacity, near here.

Ia., Washington—The city, S. J. Kelloff, Clk., will receive bids until Dec. 6 for the construction of waterworks improvements, including building, electrically driven pumps, piping, equipment and a 200,000 gal. elevated tank. A. L. Mullergren, 555 Gates Bldg., Kansas City, Mo., Consult. Engr.

Kan., Emporia—The Atchison, Topeka & Santa Fe R.R., Ry. Exch. Bldg., Chicago, plans improvements, including the construction of building, shops, power plant, equipment, etc., here. Estimated cost $5,000,000. H. W. Wagner, 9th and Jackson Sts., Topeka, Ch. Engr.

Kan., Russell—The city will hold an election Dec. 3 to vote $95,000 bonds for waterworks plant, equipment and distribution system.

La., Gibsland—The city voted $50,000 bonds for the construction of a waterworks plant, distribution system and electric light plant. E. T. Archer, New England Bldg., Kansas City, Mo., Engr. The awarding of contracts has been postponed indefinitely. Noted Oct. 30.

Md., Baltimore — J. E. Sperry, Archt., Calvert Bldg., will receive bids until Nov. 28 for the construction of a 2 story, 139 x 152 ft. power house for the Johns Hopkins Hospital, Wolfe and Monument Sts. Estimated cost $150,000. H. F. Doeleman, 507 North Charles St., Engr. Archer & Allen, Archts., 1 East Lexington St., will receive bids until Dec. 5 for the construction of an 8 story school of hygiene building for the same owner. Estimated cost $1,000,000. Heating and ventilating system will be installed.

Mich., Adrian—The city will hold an election Dec. 11 to vote $595.000 for sewer improvements, including construction of a storm and sanitary sewer, $318,430; sewage treatment plant, pumping station and main, $147,000; sanitary sewer along Raison River, $116,000.

Mich., Cadillac — The Consumers Power Co., Jackson, is having plans prepared for the construction of an earth dam and power house for 62 ft. head on the Manistee River, near here. Quotations are desired. E. M. Burd, Engr.

Minn., Willmar—The city, H. Gunderson, Clk., is having plans prepared for a complete sanitary and storm sewer system, to include remodeling present sewer system and installing new treatment plant with pump, filter beds, etc. Estimated cost $100,000. Claussen & Carroll, 406 Metropolitan Bank Bldg., St. Paul, Engrs.

Miss., Clarksdale—Burkes Landing Drainage Dist., c/o Secretary County Bd. of Supervisors, will soon receive bids for the construction of a pumping plant and drainage ditches. Estimated cost $25,000.

Mo., Booneville—The city, C. H. Miller, Clk., plans to purchase a 350 g.p.m. motor driven pump for waterworks.

Mo., Carthage—The St. Louis Conference of the Methodist Episcopal Church, c/o Ozark Wesleyan College, Marion, is having preliminary plans prepared for the construction of a 2 and 3 story college, here, to supplant the present institution at Marion. Estimated cost $1,500,000. Private plans.

Mo., Desloge—The St. Louis Smelting & Refining Co., International Life Bldg., St. Louis, is having plans prepared for rebuilding its lead smelting plant, including crusher house, roll plant, power house, ore storage bins, convevor houses, etc., which was recently destroyed by fire, here. Estimated cost $200,000. Private plans.

Mo., Webster Groves (St. Louis P. O.)—The T. P. Barnett Co., Archts., Arcade Bldg., St. Louis, will receive bids about Dec. 10 for the construction of 2 story seminary buildings, including an administration building, two dormitories, dining hall, gymnasium, boiler house, etc., here, on Lockwood St., for the Eden Seminary, 6700 Easton St., St. Louis. Estimated cost $1,000,000. Equipment detail not reported. Chapel and library buildings will be built later.

N. J., Clifton (Passaic P. O.)—The Passaic Valley Sewerage Comrs., J. H. Quigg, Secy., 900 Chamber of Commerce Bldg., 24 Branford Pl., Newark, will receive bids until Dec. 4 for pumping machinery and equipment for Yantacaw pumping station, here. Former bids rejected.

N. Y., Binghamton—C. R. Stewart, 306 Security Bldg., is in the market for a 50 hp. locomotive type boiler and engine, mounted.

N. Y., Brooklyn—The Superintendent of the Bd. Educ., 500 Park Ave., New York, will receive bids until Dec. 10 for the construction of the James Madison High School, including steam heating system, on Bedford Ave. from Ave. P to Quentin Rd., here. Estimated cost $3,000,000. W. H. Gompert, Flatbush Ave. Extension and Concord St., Brooklyn, Engr. and Archt. Noted Sept. 25.

N. Y., Hamburg—The Village Bd. is in the market for motors, meter department machinery, deep well pumps and equipment. Estimated cost $8,000.

N. Y., Jamaica—The Bd. Educ., 500 Park Ave., New York, is having plans prepared for the construction of a high school on Grand Ave., here. Estimated cost $1,000,000. W. H. Gompert, Flatbush Ave. Extension and Concord St., Engr. and Archt. Equipment detail not reported.

N. Y., Newport—The Kanata Realty Co., Utica, has purchased the Newport Electric Light & Power Co., here, and plans enlargement of plant and purchase of equipment.

N. Y., New York—E. Margolies, 10 East 33rd St., is having sketches made for the construction of a 12 story loft building on West 47th St. Estimated cost $2,000,000. Cross & Cross, 385 Madison Ave., New York, Engrs. and Archts.

N. Y., Oswego — The city has approved $350,000 bonds for the construction of a 7,500 hp. electric light and power plant.

N. Y., Schenectady — The Chamber of Commerce is having plans prepared for the construction of a 14 story hotel. Estimated cost $1,000,000. H. L. Stevens & Co., 522 5th Ave., New York, Archts. Amer. Hotel Co., subsidiary of United Hotels Co., 25 West 45th St., New York, lessee. Equipment detail not reported.

N. Y., Utica—The Scala Packing Co., 713 Bleeker St., is in the market for refrigeration machinery and equipment for proposed plant.

N. Y., Watertown—The Northern New York Utilities, Inc., 58 Public Sq., plans to build an electric plant and generating station and to install a 5,000 kw. turbine. Estimated cost $150,000. Engineer or architect not announced.

N. C., Durham—The Citizens Hotel Co., J. A. Buchanan, Chn., is having plans prepared and will receive bids about Dec. 15 for the construction of an 8 and 12 story hotel on Market and Parrish Sts. Estimated cost $1,000,000. S. H. Johnson, Lynchburg, Va., Archt. Equipment detail not reported. Noted May 15.

N. C., Laurinburg—The Mayor and Bd. of Town Comrs., C. M. Fetter, Clk., will receive bids until Dec. 18 for waterworks improvements, including construction of pumping station, filter house, filter tubs, auxiliary station, concrete coagulating basin, concrete storage reservoir, steel wash water tank and tower, c.i. pipe line, 3½ mi. 6,600 volt transmission line, motor driven centrifugal pumps, accessories and filter plant equipment. G. C. White Co., Durham, Engrs.

N. C., Raleigh—The North Carolina State College of Agriculture and Engineering is having plans prepared for the enlargement of textile building and the construction of a steam power plant, about 1,000 hp., to heat college buildings and operate a 250 kva. turbo-generator. J. E. Sirrine & Co., South Main St., Greenville, S. C., Engrs.

Ohio, Cincinnati—The Cincinnati Conservatory of Music, Highland Ave. and Oak Mt. Auburn St., had plans prepared for the construction of six buildings. Estimated cost $3,000,000. W. B. Tuthill, 185 Madison Ave., New York, Engr. and Archt. Equipment detail not reported.

Ohio, Cincinnati—The Union Gas & Electric Co., 4th and Plum Sts., W. W. Freeman, Pres., plans to build a power plant. Estimated cost $15,000,000.

Ohio, Cleveland—The Federal Fdry. Supply Co., 2639 East 79th St., is in the market for a 75 hp. motor, 220 volt, 60 cycle, a.c.

Ohio, Toledo—The city voted $490,000 bonds for the construction of sewage pumping station, disposal plant and completion of intercepting sewer system, etc., to eliminate pollution of water. A. Jackson, City Bldg., Engr.

Ohio, Toledo—The City Service Dir. will receive bids until Dec. 4 for contract 62, furnishing, delivering and erecting synchronous motor driven 35,000,000 g.p.d. centrifugal pump against 78 and 87 ft. head, with accessories at Low Service Pumping Station, Toledo Filtration Plant. Estimated cost $18,600. W. G. Clark, 1046 Spitzer Bldg., Consult. Engr.

Ore., Newport—The city, G. Blanchard, Clk., plans to purchase a 15 to 25 hp. electric motor or an internal combustion engine, also a pump, 150 to 200 g.p.m., against a 200 ft. head. P. W. Beasley, 603 Ry. Exch. Bldg., Portland, Engr.

Pa., Monaca—F. Cancillo plans to build a warehouse for fruit and vegetables, to replace the one which was recently destroyed by fire. Estimated cost $40,000. Engineer or architect not announced. The owner is in the market for refrigeration equipment.

Tenn., Nashville—The city, H. E. Howse, Mayor, is having preliminary plans prepared for the construction of a 3 story city hall and market house, including a complete refrigeration system, on the Public Sq. Estimated cost $800,000. Marr & Holman, Stahlman Bldg., Nashville, Archts.

Tenn., Shelbyville—The Public Light & Power Co., L. W. Hope, Purch. Agt., is in the market for turbines, generators and electrical equipment for proposed hydro-electric plant. Noted Oct. 30.

Tenn., Union City—The city voted $125,000 bonds for extensions to sewer system, water and electric light plants.

Tex., Childress—The Childress Light & Ice Co., a subsidiary of the Texas Central Power Co., Frost Natl. Bank Bldg., San Antonio, will soon purchase a steam turbine. F. Johnson, Ch. Engr.

Tex., Houston—Harris County, Com. Court, will receive bids until Dec. 3 for complete power plant equipment for ferry service at Buffalo Bayou. Cost will exceed $5,000. Howe & Wise, 1st Natl. Bank Bldg., Houston, Engrs.

Utah, Salt Lake City—C. A. Quigley, c/o Walker & Eisen, Archts., Finance Bldg., Los Angeles, is having plans prepared for the construction of a 14 story store and office building on Exchange Pl., here. Estimated cost $1,000,000. Equipment detail not reported.

Va., City Point—The Wilson Hock Co. is in the market for a 44 x 102 in. upright boiler and a 7 x 10 in. triplex stock pump.

Va., Richmond — The Amer. Glass Co., Broad and Meadow Sts., is in the market for a 100 hp. oil engine.

Wash., Tenino — The Skookum Lumber Co. has applied to the Supervisor of Hydraulics fo rpermission to use the waters of the Skookumchuck River for power development. It is planned to build a dam and impound a reservoir of about 220 acre ft.

Wash., Westport—The Superintendent of Lighthouses, Custom House, Portland, Ore., plans to build a power station and signal tower station at Gray's Harbor Light Sta., here. Cost will exceed $25,000.

W. Va., Morgantown—The Preston Co. Light & Power Co., H. C. Greer, Dir., plans to build a light and power plant and will receive bids about Jan. 1, 1924. Estimated cost $100,000. Engineer or architect not announced.

Wis., Bowler—The Bowler Lumber Co., W. O. Baum, Pres., is in the market for blowers, two 165 hp. boilers, etc., for proposed $40,000 sawmill. to replace fire loss.

Wis., Janesville—The Wisconsin Utilities Co., Monroe, is having plans prepared to construct a 66,000 volt transmission line from Monroe to Oxfordville, to increase voltage of present line from here to Oxfordville from 33,000 to 66,000 volts and to increase the capacity of substation, here. Estimated cost $100,000. Private plans.

Wis., Manitowoc—The Holy Family Hospital, c/o Sisters of Christian Charity, is having preliminary plans prepared for the construction of a 2 story, 52 x 100 ft. boiler house and service building. Estimated cost $60.000. E. Brielmaier & Sons, 432 Broadway, Milwaukee, Archts. The owner is in the market for two high pressure boilers and engine.

Ont., Mildmay—The Mildmay Electric Co. plans extensions and is in the market for transformers and equipment. D. McKerracher, Walkerton Engr.

Ont., Windsor—The Empire State Ice Co., Michigan Central Yards, plans to install three ice making machines, electric motors and special refrigeration equipment, brine pumps for ice plant to supply refrigerator cars. Estimated cost $100,000.

CONTRACTS AWARDED

Ala., Fort Payne—The Fort Payne Light & Power Co., A. A. Miller, owner, is building a power dam, 109 ft. high, to develop 1,500 hp. at De Soto Falls on the Little River, near here. When this dam is completed this company will be generating electricity from three distinct power units, steam, water and oil plants.

Calif., Los Angeles—G. W. Cowan et al., awarded the contract for the construction of a 14 story, 140 x 200 ft. apartment house on North Sycamore Ave. to Hennessy Bros. & Co., 810 South Wabash Ave., Chicago. Estimated cost $1,200,000. Equipment detail not reported. Noted Sept. 4.

Ill., Chicago—The West Side Ice Co., 839 North Hoyne Ave., awarded the contract for the construction of a 1 and 2 story, 100 x 125 ft. ice plant on Hoyne Ave. and Walton Pl. to the E. W. Sproul Co.. 2001 West Pershing Ave. Noted Nov. 6.

Ill., McLeansboro—The city awarded the general contract for pipe, special castings, 400,000 g.p.d. filter plant, etc., to Nixon & Keeley, Murphy Bldg., East St. Louis; pumping equipment to the Gould Pump Co., Syndicate Trust Bldg., St. Louis, Mo. Noted Oct. 23.

Kan., Kansas City — The city awarded the contract for the construction of a 259 x 433 ft. reservoir, 24 ft. high to the Arkansas Bridge Co., 1503 Waldheim Bldg., Kansas City, Mo., $274,600; two 600 hp. boilers and one 12,500 kva. turbine generator to the Babcock & Wilcox Co., 140 South Dearborn St., Chicago, $40,320.

Miss., Moorhead—The city awarded the contract for waterworks plant, including a 100,000 gal. tank, pipe lines and two 500 g.p.m. pumps, to G. P. Sledge, Abbeville, La. Estimated cost $50,000. Noted Oct. 23.

N. Y., Brooklyn — The Bd. Educ., 500 Park Ave., New York, awarded the contract for the construction P. S. 210, 5 story, 170 x 193 ft., on Rochester Ave. and Park Pl., here, to the Turner Constr. Co., 244 Madison Ave., New York. Equipment detail not reported. Noted Mar. 27.

N. Y., Vischer Ferry—Dept. Pub. Wks., Bureau of Canals, R. K. Fuller, Comr., Albany, awarded Contract B-4 for the construction of superstructure of power house at the dam, here, to the McCann Bldg. Co., 82 State St., Albany, $143,567. Noted Nov. 20.

Tenn., Memphis — The Gayoso Lumber Co., Florida and Fay Sts., will build additions to its plant, including boiler house, dry kilns, installation of boilers and equipment, by day labor.

Ont., Trent River — The Hydro-Electric Power Comn. of Ontario, 190 University Ave., Toronto, awarded the contract for the excavation of rock in tailrace of power development at Dam 9, here, to the Lumsden Eng. & Transport Co., 6 Adelaide St., E., Toronto. Estimated cost $75,000. F. A. Gaby, Ch. Engr.